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**PRINCIPLES OF
HEATING, VENTILATING AND
AIR CONDITIONING**

**WORKS OF
ARTHUR M. GREENE, JR.**

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**PUBLISHED BY
JOHN WILEY & SONS, INC.
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PRINCIPLES OF
HEATING, VENTILATING
AND
AIR CONDITIONING

BY

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NEW YORK

JOHN WILEY & SONS, INC.

LONDON: CHAPMAN & HALL, LIMITED

1936

41799

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PREFACE

This development of the principles of heating, ventilating and air conditioning has been carried out with the same purpose and in the same logical order as in the earlier textbook of the author. Since the publication of that book, the excellent experimental works of the members of the American Society of Heating and Ventilating Engineers and others at many laboratories in this country and abroad have determined many new constants and confirmed others. These data have been collected and arranged with earlier data in the present writing for use of the student and the professional engineer. For all of these proper acknowledgment is made.

The necessity for air treatment for health or comfort and for industrial or commercial purposes has led to the inclusion of air conditioning. This has required a more complete discussion, than in the previous work, of the properties of air and its treatment for the functions it performs in spaces to which conditioned air has been supplied.

As in the earlier work, the author has been guided in arranging the order of the development of the subject, for students of engineering, engineers, architects or building superintendents, by his experience as a teacher and as an engineer. On account of the lack of familiarity of many young engineers with building construction, the description of details has been covered without which the meaning of many terms would be unknown. Likewise he has included many redrawn cuts and data sheets of the products of well-known manufacturers to familiarize the reader with the devices which are used for definite performances. The commercial devices better illustrate the principles of these subjects than conventional sketches. All of them are employed to show the theoretical applications of thermodynamics developed throughout the text, and, by means of numerical examples, the author has indicated typical problems arising in practice.

The author has followed the recommendations of the American Standards Association in the omission of periods in abbreviations, and in the selection of them and the symbols for various quantities. The meaning of these symbols and their dimensions are given in the first table of the Appendix, while the equivalents of physical units are found in the fourth table of the Appendix.

Although the author has endeavored to bring all data needed for design into one volume to serve as a handbook for the engineer and architect, the work has been written as a textbook for a college course in heating, ventilating and air conditioning. It is intended for upper-class students who have concluded the course in thermodynamics. In working problems the slide rule has been used and solutions have been arranged in tabular form as suggested procedure for practice.

For more extended data reference is made to the annual Guide issued by the American Society of Heating and Ventilating Engineers. This publication contains descriptive matter of many forms of apparatus used in heating, ventilating and air conditioning.

Throughout the book, in text or footnote, the author has endeavored to give references and at the same time acknowledge his indebtedness to authors, experimenters, publications and manufacturers. To them for these aids, and especially to his wife, Mary E. Lewis Greene, for her assistance in preparing copy and in the reading of proof, he expresses his thanks.

ARTHUR M. GREENE, JR.

PRINCETON, N. J.,
June 15, 1936

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PRINCIPLES OF HEATING, VENTILATING AND AIR CONDITIONING

CHAPTER I

METHODS OF APPLICATION

Heating, ventilating and air conditioning, as practiced today, not only care for the loss or gain of heat through the windows, walls, floors, ceilings and roofs of buildings and for the supply of fresh air needed on account of physical and process contamination, but they also care for the temperature, humidity and cleanness of the air and for its movement. Heating primarily refers to the supply of heat made necessary by the losses or gains through walls and roof, by air leakage and by operations within the space to be heated. Ventilating refers to the movement of air within the spaces to be ventilated, including its supply and removal at proper places with the necessary addition of fresh air, and the cleaning of recirculated air. Air conditioning refers to the proper movement of air and its treatment to maintain definite air conditions within given spaces in which there may be sources and absorbers of heat and moisture as well as sources of contamination.

The purpose for which the room is used determines the quality and quantity of air supply: its temperature, humidity, purity, velocity and volume. The structure and size of the room, its windows, walls, floor and ceiling, and the temperatures and exposures on the two sides of each, fix the heat loss of these parts. The quality of air in the various parts of the system and the use of the room fix other quantities of heat. From the data mentioned, calculations establish the heat and air volume needed, and the engineer must arrange the various parts of the systems so that each will function to produce its own portion of the complete project. Some of the methods of supplying this heat are indicated in the figures of this chapter, in which a small building is shown so as to include all parts from foundation to roof. Although a small building is chosen, the principles illustrated may be applied to the largest structure.

The amount of air supplied to a room may be fixed by the building construction itself or by the requirements established by the use of the

room. Provisions must be made to introduce the desired quantity by the arrangement of structures or by the installation of various devices. The air itself may require treatment before it is admitted to the space to be ventilated on account of the demands of the service attempted.

For the heating and ventilating of residences or small buildings, the warm air furnace system is often used as it is less expensive than other systems of heating and is quite as efficient when proper furnaces are installed in a correct manner. An arrangement of such is indicated in

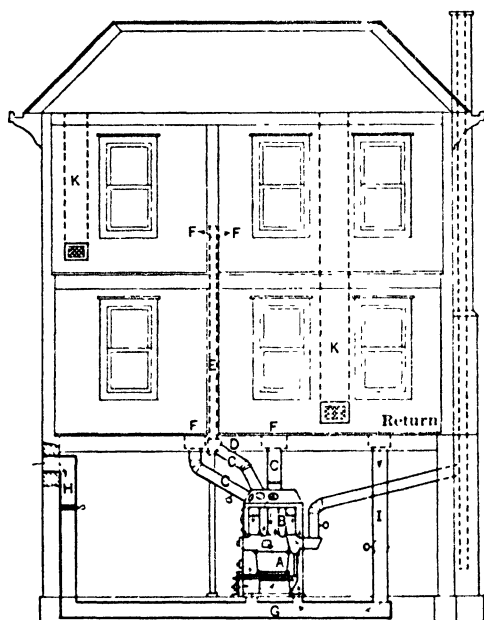


FIG. 1.—Warm-Air Furnace Heating.

Fig. 1. Heat from the fuel in the firepot, *A*, and from the hot gases in the radiator, *B*, is transmitted to the building air on the other sides of these parts by conduction and radiation. This air enters the space, *G*, below the furnace from the fresh air inlet, *H*, or from the recirculating riser, *I*. After this is heated, it leaves the top or dome of the furnace through the leader pipes, *C*, and passes into the risers or heat stacks, *E*, through the boots, *D*, rising because of the chimney effect produced by this hot air in the risers. The difference in weights of a given height of hot air and room air produces this forcing pressure or head. The air enters the room through register faces or grilles, *F*, which are located at the floor level. The register faces are controlled by damper slats. Dampers control the relative amount of air from the fresh air supply and from the return or recirculating riser. The return riser contains a grille or register at inlet, placed in the floor or in a side wall panel at the first floor level. The temperature of the hot air is fixed by the use of the room, and the amount is such that, when this quantity is reduced from the temperature of discharge to the desired room temperature, the heat loss requirements for the room have been cared for by this cooling.

As will be seen later, the furnace may be constructed to use gas or

oil as a fuel, and, if power is cheap, even electrical heating units might be used at the bottom of each heat stack.

In some furnace heating systems no air is recirculated, the foul air from the rooms being taken off through vent stacks, *K*, leading to an exhaust duct, and all air for furnace heating is taken in from the outside through *H*. Such a system is employed only if large numbers of persons are present in the building or if the air becomes contaminated as the result of some process carried on in it. In ordinary residence heat-

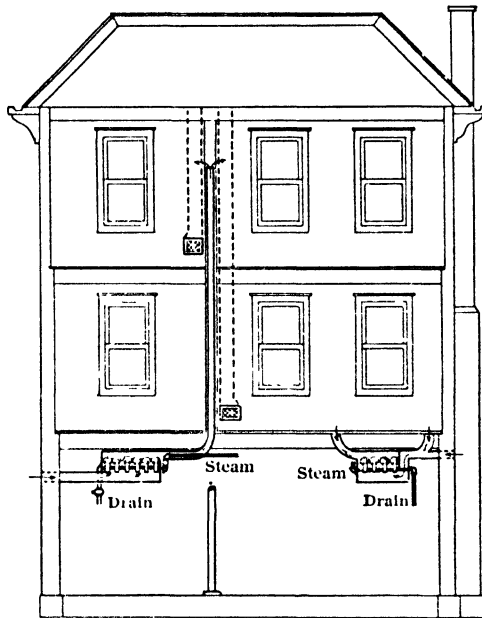


FIG. 2.—Indirect System of Heating with Individual Stacks.

ing the leakage through windows and doors supplies more than sufficient fresh air, so that all air for heating may be recirculated from the house.

If the ground area of a building is great the small pressure produced by temperature differences of air in parts of the system is too slight to cause proper circulation and it is necessary to place a fan in the duct, *G*, or beyond the heater to give the necessary pressure head for remote circulation.

If the warm air method of heating is desired in large buildings, indirect steam radiators may be placed in the cellar at the bottom of each riser or below each floor register of the first floor to heat the required amount of air to the proper temperature, as shown in Fig. 2. The steam

or hot water is carried from the boiler or water heater to these indirect radiators by pipes of proper size and the fluid flow is possible, as sufficient pressure difference may be developed easily for remote circulation.

In some buildings (see Fig. 3), all indirect radiators, *D*, are concentrated at one point for heating the air, and a fan, *C*, produces sufficient pressure to circulate the air through riser flues, *A*, and ducts, *B*, to the remotest part. The system in which pressure, produced by the fan, drives the air into the building is known as the plenum or pressure system. Although Fig. 3 is drawn as a residence, the same system may

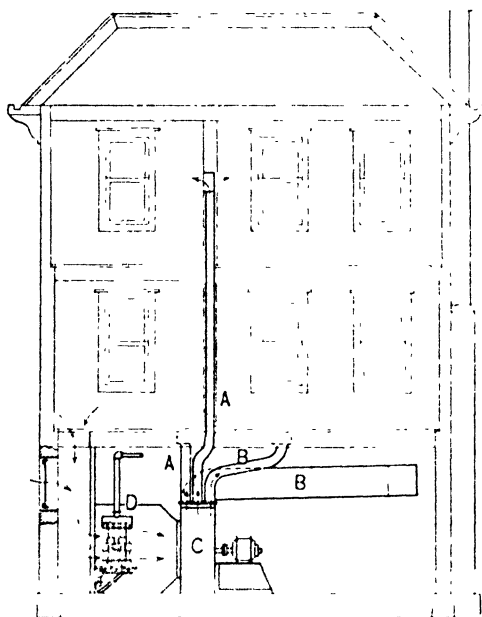
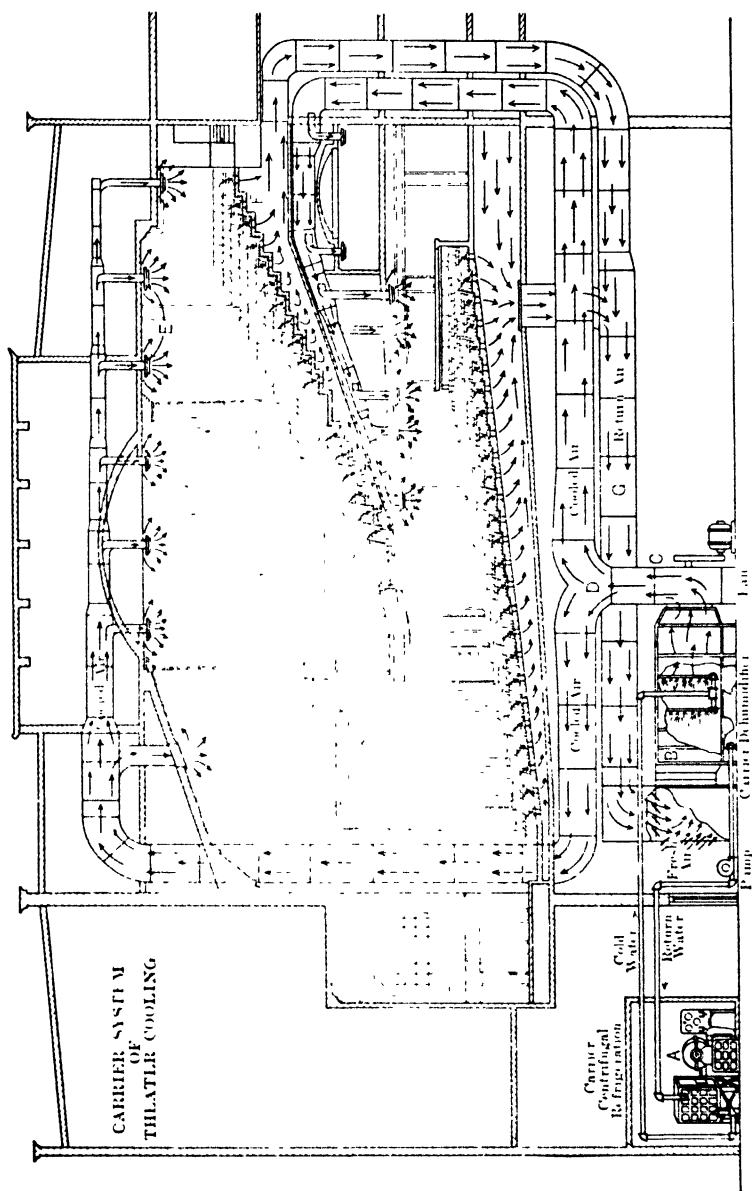


FIG. 3.—Plenum System of Indirect Heating with Single Unit.

be used for theatres, Fig. 4, for auditoriums, for office buildings or for factories, Fig. 5.

In the vacuum system of ventilating and indirect heating, an induced draft fan is used to remove foul air from the duct system of the outlets from the rooms.

These positive systems of air circulation are applied whenever the occupation or use of the building demands a definite supply of air to each room. If only a limited amount of fresh air is required, the regulation of the fresh air inlets of the plenum fan or of the free outlet of the vacuum fan will give the desired results, the recirculated air making up the difference for the total needed.



Courtesy of Carrier Engineering Corporation

FIG. 4.—System of Indirect Heating and Air Conditioning.

Where steam or hot water is employed in radiators placed in each room, such an arrangement as that shown in Fig. 6 is called a direct system of radiation. In this the boiler (or hot water heater), *A*, supplies steam through distributing main, *B* (called flow main for water), to the various steam risers, *C*, and thence to the radiators, *D*, by the run-outs or branch radiator connections, *E*. The condensed steam (or cooled water), called the return, is

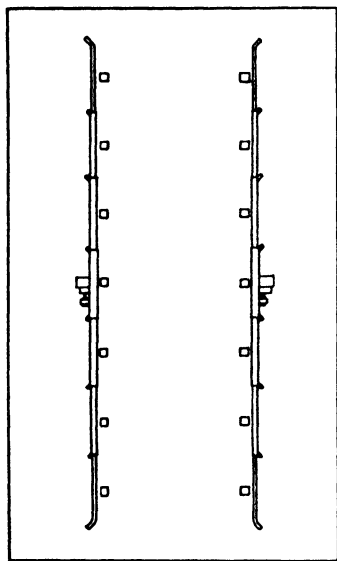
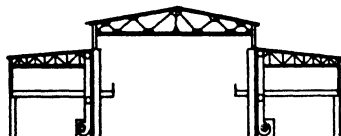


FIG. 5.—Indirect Heating of Shop Building.

carried through the branches, *F*, the return risers, *G*, and the return main (flow line for water), *H*, to the boiler or heater, *A*. The radiators are often placed beneath windows as the upward currents of warmed air form blankets through which the cold leakage air does not pass directly into the room and produce drafts. Moreover, this window space is not occupied by furniture, and placing radiators in front of the windows leaves other wall space available for furniture. The modern method of placing radiators within recess cabinets beneath the window sills or in the walls, with grilles for the inlet and outlet of air, permits both wall space and window space to be free from projecting radiators. This use of concealed radiators in recesses below the window sills is quite common when projecting radiators are to be eliminated.

Each radiator is controlled by valves at the inlet and outlet. The risers are shown exposed in the rooms, but often they are placed in wall

chases so that pipes also are concealed. With such concealment an endeavor should be made to provide for repairs when needed.

As shown, no ventilation is cared for specifically in Fig. 6. The air leakage through the windows and walls gives the necessary fresh air for a small number of occupants, and the radiators are sufficiently large to heat this leakage air as well as make up for heat loss through the walls. When ventilating air must be supplied on account of large occupancy, air, tempered to temperatures of the rooms, may be intro-

duced under pressure at some point in each room. This system of direct radiation to care for heat loss with tempered air forced in for ventilation is called the split system of heating and ventilating. It is advantageous as the fan may be shut down during periods of non-occupancy and the proper temperature will be maintained by the radiators.

In all the above systems a central unit or heater supplies heat. The heat originates from some fuel: wood, coal, gas or oil, or it may be sup-

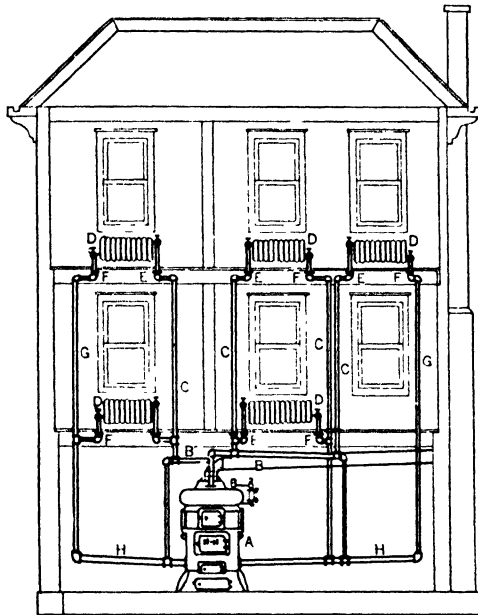


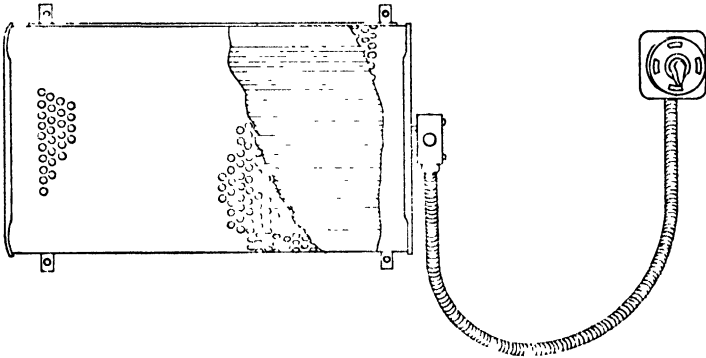
FIG. 6.—Direct Heating by Steam or Hot Water.

plied by an electric heating unit. Individual electric heating units have been used for many years when heat is needed for a short time in some rooms. It may be the main source of heat if electric energy can be bought at a low rate. One form of electric heating element, Fig. 7, delivers the heat directly to the room; in Fig. 8 a unit is shown in which the energy from the electric element volatilizes a liquid, the vapor from which is used in the radiator of extended heating area to furnish heat for the room through radiation and convection from a larger surface within a cabinet.

To compare the cost of energy for heating, a table is given below in which the number of Btu for heating obtained from fuel or energy

costing one cent is listed for various sources of energy. These have been computed as follows:

$$\frac{(\text{Btu per unit})(\text{Efficiency})}{\text{Cost of one unit in cents}} = \text{Btu delivered for one cent of energy or fuel cost}$$



Courtesy General Electric Co

FIG 7. Unit Electric Heater.

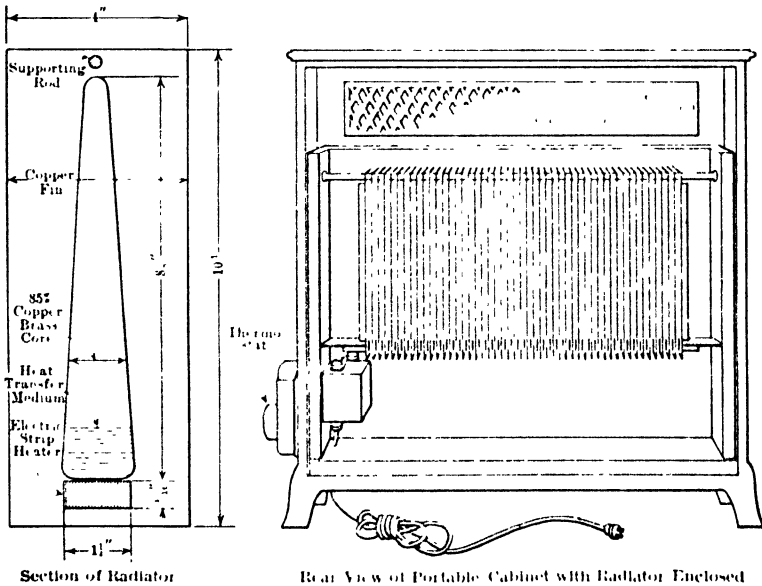


FIG. 8.—Chelva Electric Heating Cabinet.

Of course, the cost of attendance must be considered in addition to the fuel cost. Changes may be made in the table for other costs of

the fuels or electric energy. Although electric energy is very expensive even at one cent per kilowatt-hour, the concentration of this energy and its complete conversion together with its simplicity of application

TABLE 1-1
HEAT OF ENERGY SOURCES FOR ONE CENT

Supply	Btu for energy cost of 1 cent	Attendance
a. Wood @ \$8.00 per cord (60% efficiency)	17 100	Considerable
b. Anthracite egg @ \$10.00 per ton (60%)	15 100	Appreciable
c. Anthracite buckwheat @ \$4.50 per ton (60%).....	30 000	Slight; energy for blower required
ca. Anthracite buckwheat @ \$4.50 per ton in central station (75%)....	38 800	Appreciable energy for blower required
d. Bituminous coal at \$4.00 per ton in cen- tral station (80%)	55 000	Appreciable energy for blower required
e. Manufactured gas @ 75¢ per M (75%)	5 000	Slight
f. Natural gas @ 25¢ per M (75%)	30 000	Slight
g. Fuel oil @ 5¢ per gallon (80%) . . .	23 400	Slight, energy for blower required
h. Electric energy @ 1¢ per kw-hr (100%)	3 414	Inappreciable

where needed make it quite advantageous for short periods of local heating.

The air in the warm air furnace system, Fig. 1, is introduced near the floor on one side of a room, and exhaust air should leave the room, if possible, at a point near the floor on the other side of the room. The endeavor should be made to cause the hot air, which has a very low velocity in this system, to mix with the air in the room before it is carried away through the outlet duct registers or through the doors to a central recirculating register on the first floor. Although there are some advantages for distribution through the room in bringing the hot air up through a duct or riser in the outer wall, it is thought that heat loss in such locations is a bad feature, and leaders should be placed in inside partition walls as shown in Fig. 1. The fresh air which may enter on the windward side of such buildings through window cracks causes a like discharge through vent stacks or through window cracks on the leeward side of the building.

In Fig. 3 the plenum system is indicated. The air may be introduced into the rooms in two ways. If introduced through a stack or flue in an inner wall of the room, Fig. 9, it should be discharged above

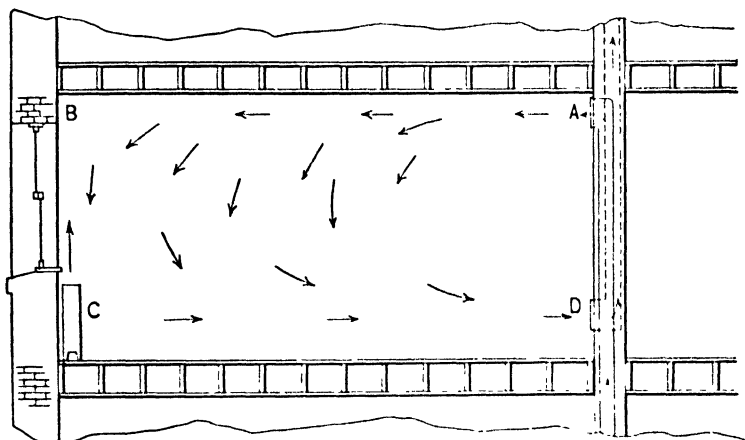


FIG. 9.—Method of Circulating Air.

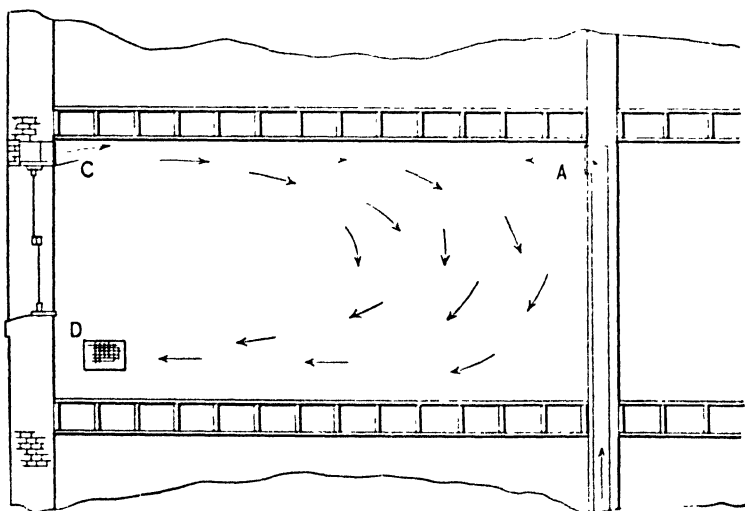


FIG. 10.—Method of Circulating Air.

head height, preferably near ceiling, and delivered toward one of the outer walls. This causes the room air to mix with the warm air, and as it reaches the floor in its downward course at the opposite wall, the removal of the air at the floor line on the original side, as at D, will

mean complete circulation. If the air enters at *A*, Fig. 9, and is removed at *B* or *C*, there is a chance that the ventilating or heating air will be carried off before it is effective.

If the air is introduced from the outer wall of the room as at *C*, Fig. 10, it should be directed upward so as to mix with the room air by diffusion before it is removed at *D*. The admission of air at *A*, Fig. 10, would prevent the mixing of the supply and room air on its way from *A* to *D* so that the full value of the air for ventilation or heat would not be obtained.

In the split system the heat from a radiator on the outer wall side of the room will produce an up current which mixes with the down current from *B* to *C* of Fig. 9. If the outlet were at *A* instead of *D*, Fig. 9 or Fig. 10, the up currents from the radiator or the discharge of air from *C* would go directly to the outlet, short-circuiting the room and reducing the heat received from the air.

The outlet should always be at the floor level unless rapid clearing of the room is desired, in which event floor and ceiling registers are placed on the same vent flue, the ceiling register being used occasionally.

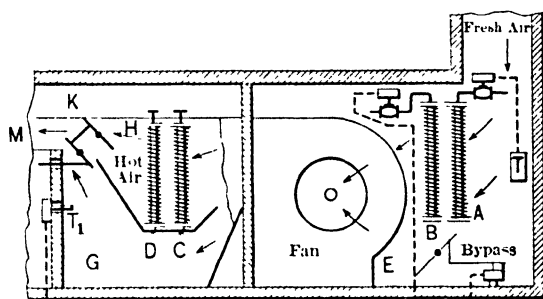


FIG. 11.—Aerofin Heaters for Plenum System.

Air is heated in the plenum system of heating by means of heating units *D*, *C*, Fig. 11, supplied with steam, hot water or some other source of heat energy, the air being blown over the heating units by a fan. The air, sucked from the chamber *E* by the fan, enters the space *E* from the outside, passing over tempering coils *A*, *B*, which warm the air to the desired temperature for ventilation, the higher temperature for heat loss being obtained by the heating units *D*, *C*. Thus, the air in *G* is of the same temperature as that in *E*, while the air in *H* is at a higher temperature. The dampers, *K*, regulate the relative amounts of air passing from each space to give a desired temperature in the outlet duct *M*. A by-pass damper below the tempering coils *A*, *B*, and controlled by thermostat, *T*₁, furnishes means of rapidly controlling the temperature in spaces *E* and *G* before the steam supply can be regulated in *B*. When recirculation is desired the fresh air inlet is connected to the return duct from the building with dampers in the fresh air inlet and in the return duct. These are placed in advance of the

point of meeting so as to regulate the relative amounts of fresh and return air. The dampers of these ducts are shown in Fig. 4.

With the split system, the air introduced is for ventilation only and it is at one temperature for all parts of the building, as the direct radiation cares for the varying amounts of heat for different rooms. The fan of Fig. 11 would then deliver directly into the outlet ducts but the by-pass damper and the thermostats T and T_1 would be used.

If there is not sufficient direct radiation in the rooms to supply the heat loss, or the system is fully indirect, the air introduced at the grille must be at a temperature above that of the room. The temperatures of this air and the amount of air are such that, when the air is reduced to the desired room temperature, the heat given up in cooling will just equal the heat demand for the room. Since the air quantities for ventilating are different for various rooms and since the heat requirements usually do not vary in the same way, it is seen that the temperatures of the entering air for various rooms will be different, and of these temperatures there will be one of the highest and one of the lowest value. In general, then, it will be necessary to have part of the air heated to the highest temperature needed and also some air heated to the lowest temperature required, and, by mixing quantities of air at these two temperatures, the other supply temperatures can be obtained. The reason for the two temperatures at G and H of Fig. 11 is evident.

The mixing of the air to obtain the desired temperature may occur near the heater, as at K , Fig. 11, and from this point a single duct is taken to each room, forming the single-duct system. The full discharge from chambers G and H may be carried by two ducts which extend around the building, and at different points of these ducts connections are made to single flues leading to different rooms, the temperature of the air mixture being controlled by a damper or splitter. This forms the double-duct system. In both single-duct and double-duct systems there are single risers or flues carrying air at the proper temperature to a given room, but in the first there are a great many ducts leading from the heating chamber, one to each room, and in the second only two ducts to which a number of individual risers are attached at selected points.

To mix the air, the damper at K , Fig. 11, diminishes the air volume at one temperature while increasing it at the other in the single-duct system. In Fig. 12, the mixing damper of a double-duct system is shown.

Air movement must take place in any space for proper ventilation, as this motion prevents the formation of a warm wet blanket of stagnant air around the occupant. Although this motion is needed, it must not be so great as to cause the sensation of drafts, and the American Society

of Heating and Ventilating Engineers have set 50 feet per minute as the maximum velocity of air currents measured at a height of 36 inches from the floor when the only sources of air contamination are the occupants of the room. The velocity of the air currents within this maximum limit must be such as to maintain a reasonable uniformity of temperature and humidity throughout the room.

As a velocity of 400 to 800 fpm (feet per minute) is used in vertical flues, while the discharge through ordinary room registers or grilles is at 200 to 400 fpm, it is evident that the velocity of the air in the flue would be continued on into the grille and room, as shown in Fig. 13, even though there be a gradual enlargement from flue into the boot behind the grille. The

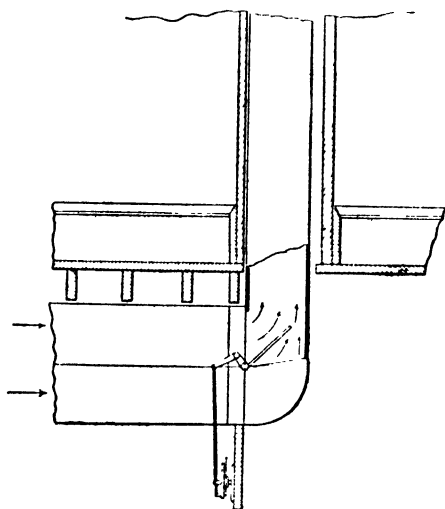


FIG. 12. -Mixing Damper Thermostatic Control. Double Duct System.

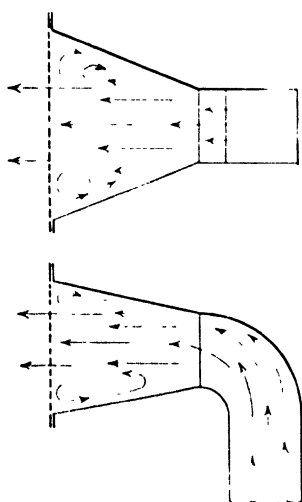


FIG. 13. -Unequal Discharge.

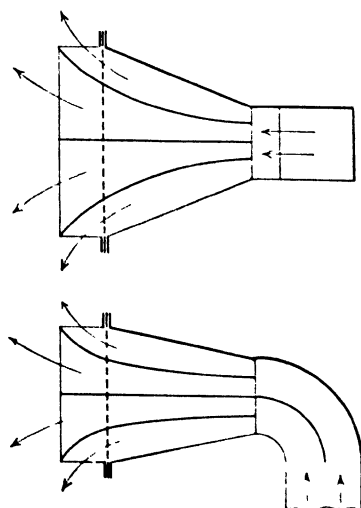
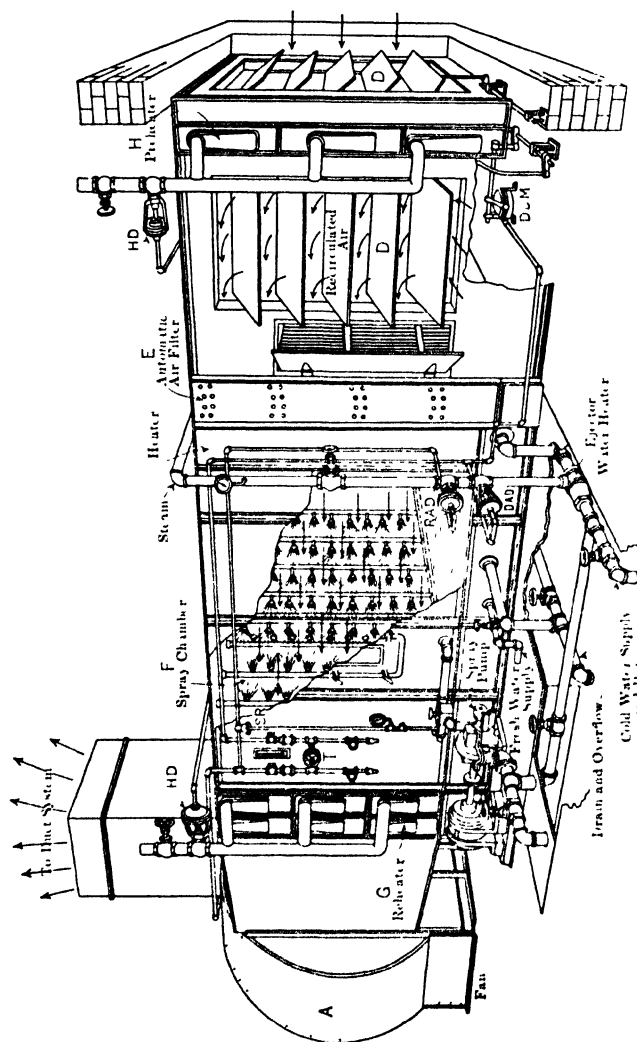


FIG. 14. -Splitters for Equalizing Discharge.

velocity of discharge from the grille would then produce lanes of dis-

charged air into the room. To prevent this, vanes or splitters are introduced into the elbow behind the grille and also over the face of the grille, as shown in Fig. 14. The vanes over the grille may be arranged with vertical or horizontal elements, or a combination might



Courtesy of Carrier Engineering Corporation.
FIG. 15.—Air Washer and Conditioner.

be used. The purpose of these vanes is to produce a uniform discharge at a much lower velocity than that found in the flue or at the grille.

The air entering the room from the flue is conditioned for tempera-

ture; humidity; toxic, unhealthful or disagreeable gases or fumes; odors; bacteria, and dust. This conditioning is accomplished by passing the air through an air washer and humidity regulator, and then over heating coils when necessary to bring the air to some desired temperature. It has been suggested that the air be ionized to give it the quality of

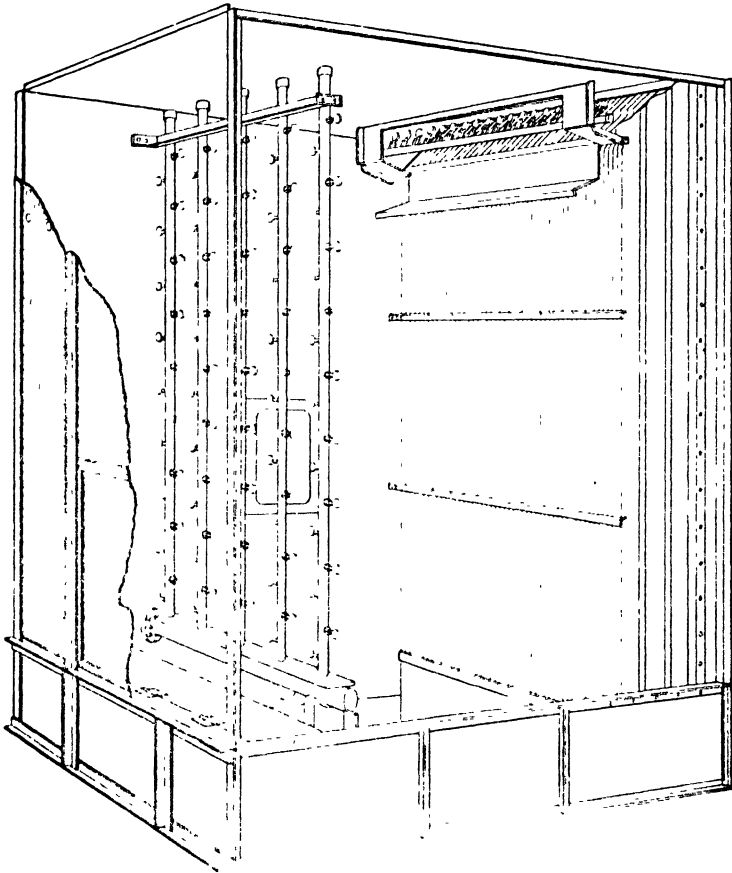


FIG. 16.—Air Washer of Buffalo Forge Company.

comfort and health found in outdoor air, although complete experimental data are not at hand regarding the advisability of this procedure.

One form of air conditioner is shown in Fig. 4 and another in Fig. 15. A fan, *A*, in the latter delivers air into a duct, drawing it through the conditioner from the outside or from the recirculation duct through the cross-connected damper shutters, *D*, the filter plates, *E*, the spray chamber, *F*, and the reheater, *G*. The air passes over a tempering coil,

next the filter, to bring the cold air nearer to the temperature of the water spray in the spray chamber. This spray is used to free the air of impurities and, by fixing its temperature, to bring it to the proper dew point for the moisture content desired in the air at outlet. In the summer time, when air is to be cooled and dehumidified, it is customary to supply cooled water to the spray heads, the temperature of which is such that after removing the excess moisture from the mixture of recirculated air and fresh air, and then cooling the mixture, the temperature of the air approaching the eliminators in front of the heating coil will be that of the dew point corresponding to the moisture content at outlet. The eliminators are a series of baffle plates for removing the drops of washing water which may remain in the air. The pump shown in front of the conditioner may recirculate the water from the basin of the

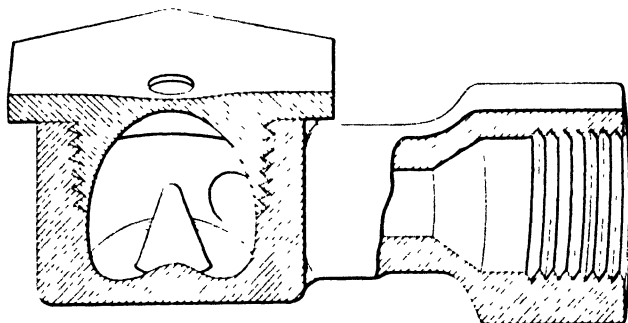


FIG. 17. Buffalo Forge Spray Nozzle.

washing chamber when the water in the basin is cooled or heated to the proper temperatures for the desired outlet condition, although usually, for summer conditioning, cold water is supplied from a separate refrigerated cooler at another point. The air, after being brought to the proper moisture content by the sprays, may be heated to a desired temperature by the heating coil or reheater, *G*. Sometimes the recirculated air may be by-passed around the washer if it is found that the air from the room is of proper purity, humidity and temperature; otherwise it is passed through the washer as shown.

The detail of a washer and humidifier is indicated in Fig. 16. Air at about 500 fpm enters from the left and passes through a spray of water of proper temperature delivered from a bank of spray nozzles. Each nozzle, as shown in Fig. 17, delivers 1.5 to 1.8 gallons of water per minute at 20 lb per sq in., on a tangent to the circular cavity, setting up a vortex which causes the water to issue with such a swirl that it breaks up into a mist projected toward the eliminator plates beyond

them. These are at a distance of over 4 feet, or at a distance of 3 feet from the rear bank when two banks are used at 3 foot centers. The eliminators shown in Fig. 18 are made up of single sheets of metal placed at $1\frac{1}{8}$ in. centers. These sheets are bent into six surfaces at angles of 120° , as shown, and each is so bent that lips are formed at the ends of the last three surfaces to catch drops of moisture and allow this water to fall. The first surfaces are flooded by a spray from the top which continuously washes the surface free from dirt particles which collect. The spray nozzles supply 4 to 10 gallons of water per thousand cubic feet of air while the flooding nozzles supply $\frac{1}{2}$ to $\frac{1}{8}$ as much. Inspection doors on the side with glass windows permit visual inspection of the

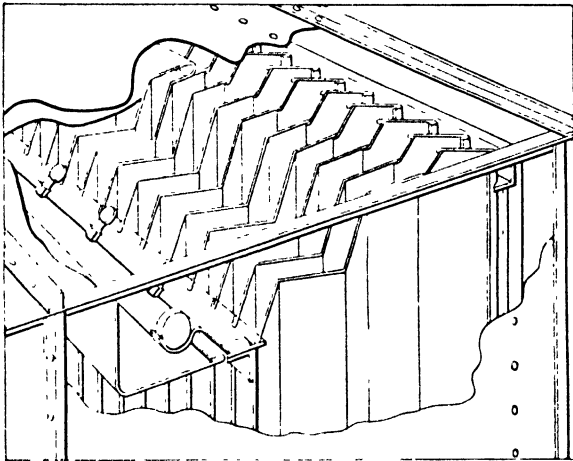
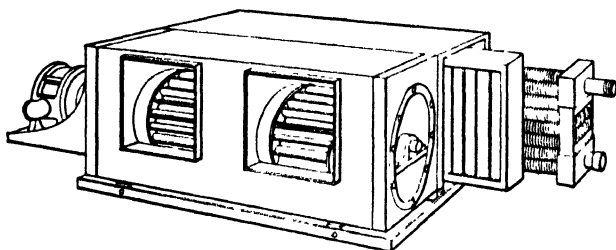


FIG. 18. Buffalo Forge Eliminator.

interior, illuminated by electric lamps, while a metal inspection door over the eliminator gives access to this. The basin at the bottom must be kept clean by renewing the water at intervals.

Unit air conditioners may be used at different points of a room. In these, a motor-driven fan at the top of the casing draws air through the base of the cabinet from the outside at one inlet and from the room at a lower inlet. The air passes through a spray chamber, the water for the spray being lifted by a high-speed pump from the reservoir at the base. After washing and eliminating drops of water, the mixture of air is warmed to the proper temperature by steam coils below the fan. This air is then delivered at high velocity from openings at the top or side of the unit into the space where the air conditions are to be maintained.

A unit cooler to be mounted on the ceiling of a room for air conditioning is shown in Fig. 19. In this the coil of copper or steel, with fins, is supplied in the summer with brine, cold water or a volatile refrigerant and cools the air, depositing the moisture on the eliminator



Courtesy of Buffalo Forge Co

FIG. 19.—Unit Suspended Air Cooler for Use near Ceiling.

plates before entering the fan space. The fan delivers the conditioned air with sufficient velocity to distribute it throughout the space to be cooled. These same units are built to stand vertically on the floor and deliver horizontally from elbows for distribution into the room. In cold weather heating fluids may be supplied to the finned radiator.

CHAPTER II

PROPERTIES AND CONDITIONING OF AIR

Air is a mixture of gases, principally nitrogen and oxygen, with small amounts of argon, carbon dioxide, hydrogen, helium, neon, krypton and xenon, and a variable amount of water vapor. The oxygen in dry air amounts to 23.2% by weight or 21.0% by volume, argon 1.3% by weight and 0.94% by volume. The remaining part is nitrogen and the other gases. If all gases except the oxygen are treated as nitrogen, the following analyses are generally given:

	By Weight	By Volume
Nitrogen.....	76.8%	79.0%
Oxygen	23.2%	21.0%

Free atmospheric air contains from 3 to 4 parts of CO₂ per 10,000 parts of air, and for many years the amount of CO₂ per 10,000 in any ventilated space was a measure of the completeness of ventilation. Experiments have shown that the air expired from the lungs which, on the average, amounts to 400 cu in. per min (20 cu in. per respiration, 16 to 24 respirations per minute), contains 4% CO₂. This gives 16 cu in. per min or 0.56 cu ft of CO₂ per hour per person as the amount added to any occupied space.

This exhalation increases the content of CO₂, and the maximum allowable amount per 10,000 parts has received much discussion. Some advocate 8 parts per 10,000 as a maximum, but experience has shown that in breweries the CO₂ contents of 50 to 200 parts per 10,000 have been maintained without injurious effects on occupants.

With 200 parts of CO₂ there is deeper breathing. This augmentation (50%) is automatic like that which occurs in moderate exercise. When the CO₂ content reaches 500 to 700 parts (5 to 7%) breathing becomes labored, and with 10 or 11% headaches, nausea and chilliness may be experienced.*

It was once thought that the amount of CO₂ was an indicator of other contamination and that, when it reached values of 10 parts in

* Report, New York State Commission on Ventilation, 1923. Chapter I is an excellent discussion of the historical views on ventilation.

10,000, the toxic effluvia and other emanations were sources of danger. Experiments have been made to check the presence of toxic material in expired air, and as indicative of the experimental data in 1913, at the beginning of their work, the New York Commission stated: "The weight of scientific evidence as based on physiological investigation was, however, against the existence of such hypothetical substances."

The amount of air supplied per person can be calculated by determining the CO₂ content of a room and of the outside air as follows:

Let V = cubic feet per hour supplied per person ;

0 = parts CO₂ per 10,000 in outside air ;

p = parts CO₂ per 10,000 in room air ;

0.56 = CO₂ produced per person per hour in cubic feet.

$$\frac{0 \times V}{10\,000} + \frac{5600}{10\,000} = \frac{p \times V}{10\,000}$$

$$(p - 0) V = 5600 = 6000 \text{ (approx.)}$$

$$V = \frac{5600}{p - 0} \quad \text{or} \quad \frac{6000}{p - 0} \text{ (approx.)} \quad (2-1)$$

For 4 parts in 10,000 to 7 parts in 10,000

$$V = 1867 \text{ cu ft per person per hour}$$

This is practically the amount of fresh air per hour per pupil (1800 cfh) required by law in many states for school buildings. Although the laws of the state must be examined in any given problem, the ventilation standard of the A.S.H.V.E. (American Society of Heating and Ventilating Engineers) requires in Section V that not less than 10 cu ft per min (cfm) of air per person from an outdoor source must be mixed with the air circulated in such quantities as to maintain the standards of temperature, air quality, air motion and air distribution. This means 600 cfh per person, which gives, with 4 parts of CO₂ in outside air, 13.33 parts per 10,000 in the ventilated spaces.

CO₂ content may be increased also by open gas flames or by industrial processes, and these must be examined in any given installation.

As gas flames give off CO₂, moisture and other gases, burning gas in a room contaminates the air and may necessitate additional air for ventilation. If a gas contains by volume 10% carbon monoxide, 2% carbon dioxide, 30% CH₄, 4% C₂H₄ and the remaining volume does not contain gases with carbon, it will be found that the burning of 1 cu ft

of this mixture will produce $\frac{1}{2}$ cu ft of CO_2 , if reduced to the initial temperature and pressure. The burning of 1 cu ft of this gas per hour requires the amount of fresh air equal to that required for one person.

The amount of CO_2 in the air is usually so slight that the ordinary gas apparatus is inadequate to determine it. A form due to Pettersson is used. The bottles *A*, *B* and *C* are immersed in a water bath to equalize the temperature. By lowering or raising the mercury bottle *D*, when the cock *F* is open,

the caustic potash solution is raised in *C* to the mark on the capillary tube below the cock *F* and then after closing *F* the tube *A* is exhausted of air through the cock *E*. The air to be tested is then drawn into *A* to the bottom of the small graduated tube *M*. While this is being done *I* and *H* are open. After closing these and bringing the mercury in *D* and *M* to the same level, the cock *E* is closed and the cock *G* is open and the position of the liquid drop at *K* is noted, as well as the level of the mercury in *M*. *G* is then shut off and *F* opened, and the air is driven over into the bottle *C* where it contacts with KOH solution on glass tubes as the solution is driven into the vessel *L*. This is repeated several times, then the air,

free of CO_2 , is drawn into *A* until the KOH fluid reaches the mark below *F*, after which *F* is closed. *G* is opened and *D* is raised or lowered until the drop *K* is at the original scale reading. This means that the pressure on the gas in *A* is the same as before. The reading on the scale *M* at this time determines the diminution of volume or the amount of CO_2 absorbed. The small tube *M* permits one to determine small absorptions, and the drop *K* gives an accurate method

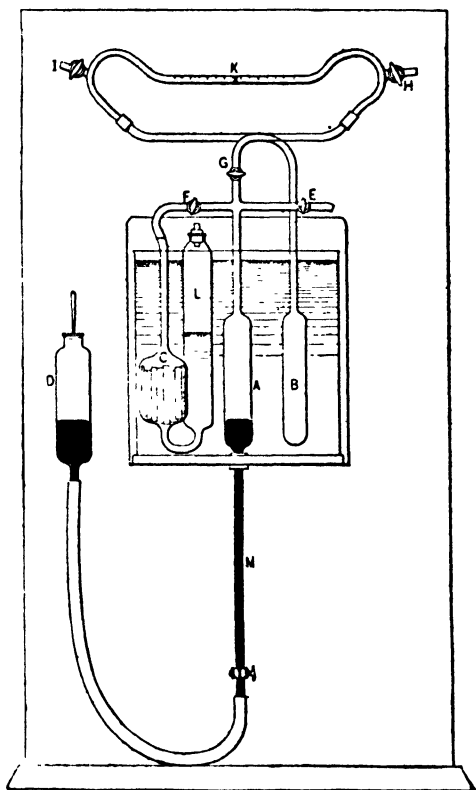


FIG. 20.—Pettersson-Palmquist Apparatus for CO_2 .

of assuring atmospheric pressure on the gas when the level of mercury in *M* is observed.

A simple apparatus for getting relative results is shown in Fig. 21. The bottom of the bottle contains a small amount of a standard solution of sodium carbonate with phenolphthalein which produces a pinkish color. If CO_2 is introduced into this until there is sufficient to change the sodium carbonate into the bicarbonate, the pink color is destroyed

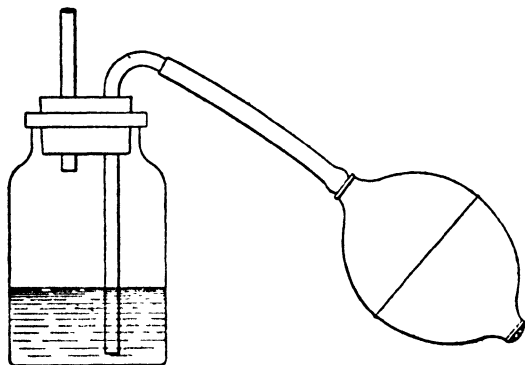


FIG. 21.— CO_2 Bottle.

and the solution becomes colorless. If, now, it takes *A* volumes of the bulb to destroy the color when atmospheric air is used, and *B* volumes for the same amount of the same liquid when taking air from a room, the number of parts per 10,000 in the air of the room will be $4A/B$. Billings gives the strength of

solution to be 5.3 grams of desiccated sodium carbonate, 1 gram of phenolphthalein and 1000 cc of distilled water which has recently been boiled and cooled. When ready to use, this is diluted still further with 25 times its volume of boiled distilled water.*

A variable component of outdoor and indoor air is the moisture or water vapor which comes from evaporation of water, from streams or other bodies of water, from rain, from living organisms and from industrial chemical reactions.

A volatile fluid, such as water, gives off molecules of vapor from its free surface into the region beyond, and this region experiences a vapor pressure due to the pressure from these molecules. The molecules of vapor in the region are moving in certain paths, and at times some of the molecules will be returned to the liquid. If the region is limited by walls, the pressure exerted by these bombarding molecules will cause the pressure of the space to increase until the number of the molecules

* The Wolpert air tester uses this principle. A graduated glass cylinder of 50 cc capacity contains a piston which draws a measured quantity of air from the atmosphere into this cylinder which contains a known amount of a soda-phenolphthalein solution. This is shaken thoroughly, and the position of the piston is varied until the pink color disappears. If 50 cc of outside air are needed to change the color, and 20 cc of inside air, the inside air will have $2\frac{1}{2}$ times as much CO_2 as the outside air, or 10 parts per 10,000.

leaving the liquid is just equal to the number returning, and then it will be found that the increase of pressure is equal to the pressure of the vapor above the liquid when it boils at the temperature of the surface of the liquid with no other vapor or gas present in this space. This is known as the vapor (steam) pressure corresponding to the temperature. Thus, for instance, if water in a vessel at 70° were maintained at this temperature by a heat supply in the bottom of a vessel, and if the space above the water were originally filled with dry air at atmospheric pressure of 14.696 lb per sq in. (written 14.696%), or 29.92 in. of mercury, and at temperature of 70° F, water would be evaporated at this temperature, increasing the pressure until the increase of pressure was 0.363 lb per sq in. (0.363%), or 0.739 in. of mercury. At this time the total pressure exerted by the mixture of air and water vapor (steam) would be 15.06%, or 30.66 in. of mercury. This is called the total pressure of the mixture, and the two individual pressures are called the partial pressures, 14.696% from the dry air and 0.363% from the water vapor. The molecules of the constituent gases of the air and water vapor are so small that each of the component parts of the mixture may be said to occupy the full volume of the space of the mixture. Dalton stated this as a law: **The total pressure of a mixture of gases (when each component occupies the total volume) is equal to the sum of the partial pressures of each component.** Another aspect of Dalton's law which should be remembered refers to separating the constituent gases into individual volumes and holding each of these volumes at the full total pressure. It is then found that when the partial volumes are added together the original total volume is obtained. This gives as the second aspect of Dalton's law:

The sum of the partial volumes of the component parts of a mixture of gases when each is under the total pressure will equal the total volume. These may be stated as:

$$p_m = p_1 + p_2 + p_3 + \dots \quad (\text{Each component occupies full } V_m) \quad (2-2)$$

$$V_m = V_1 + V_2 + V_3 + \dots \quad (\text{Each component under } p_m) \quad (2-3)$$

Since

$$p_1 V_m = W_1 R_1 T \quad \text{and} \quad p_m V_1 = W_1 R_1 T$$

from (2-2)

$$\frac{p_m V_m}{T} = W_1 R_1 + W_2 R_2 + W_3 R_3 + \dots = W_m R_m \quad (2-4)$$

from (2-3)

$$\frac{p_m V_m}{T} = W_1 R_1 + W_2 R_2 + W_3 R_3 + \dots = W_m R_m \quad (2-5)$$

Each similar term of the expressions, (2-4) and (2-5), is seen to be the same percentage of the sum. This means that the partial volume of one constituent bears the same relation to the total volume as the partial pressure of that constituent bears to the total pressure. Thus, if the partial volume of oxygen in the air is 21%, this oxygen exerts 21% of the pressure exerted by the mixture. Hence, under a pressure of 15% the oxygen accounts for 3.15% and the other gases for the remainder.

If the walls of the region above the water in the original vessel were made of a flexible membrane and held by the constant external atmospheric pressure it would be found that the wall would move outward as the water vapor was formed. The atmospheric pressure would now be supported by the air pressure and the vapor pressure. This would mean that the air would be under less pressure as the steam or vapor was formed and so the air would expand. In this case, for 70° F the total pressure would be 14.696%, the vapor pressure would be 0.363% and the air pressure would be 14.333%.

When the vapor had reached a pressure of 0.363% for 70° F, there would be no further increase as the water would receive from the space above as many molecules as it was giving out in a certain time; when the space would be said to be saturated with water vapor. No more vapor could be carried in this volume at 70° F. Should the space be cut off from the water before the saturation point had been reached, the space would not be completely saturated and the amount of moisture present would be indicated in one of two ways: either by actual moisture present per cubic foot in pounds or grains (7000 grains = 1 avoirdupois pound), called the absolute humidity; or by finding the ratio of the amount of moisture present per cubic foot to the amount required to saturate 1 cu ft of space, which is called the relative humidity. It is expressed as a percentage. Another way to express this moisture content is to give the number of pounds or grains of moisture associated with 1 lb of dry air. This is called the specific humidity.

When the vapor does not saturate a given region, its partial pressure will be below that of saturation for the temperature of the vapor and the steam is superheated. If such space is gradually cooled by the removal of heat by some external means, the temperature falls until the temperature of saturation for the amount of moisture present is reached and any further cooling will cause a precipitation of moisture; this point of saturation is called the dew point.

If air at 70° F has 0.00058 lb of water vapor in each cubic foot (4.04 grains per cu ft), this expresses the absolute humidity. Since it takes 8.07 grains to saturate 1 cu ft, the relative humidity in this example is 50%. The specific humidity of this air may be computed

to be 54 grains of vapor per lb of dry air. It is found that air at 49°F is saturated by 3.97 grains per cu ft, and at 50° there are 4.11 grains per cu ft, so that the dew point for the conditions given is slightly under 50°F .

To determine the amount of moisture in air, hygrometers of some sort are used. One method is to reduce the air or a sample of it to such a temperature that it will become saturated and deposit moisture. Fig. 22 illustrates one form of dew-point apparatus or hygrometer in which the aspiration of air through a volatile liquid in a container evaporates some of the liquid and reduces its temperature and that of its container so that moisture begins to form on the silvered surface of this at the lower end of the apparatus. The temperature at which this moisture forms can be noted and again the temperature at which the moisture just disappears. The mean gives the temperature of saturation or the dew point. The ratio of the weight of steam (or moisture) per cubic foot at this temperature taken from steam tables to the weight of a cubic foot of dry air at the partial pressure of the air gives the specific humidity, from which the relative humidity may be computed.

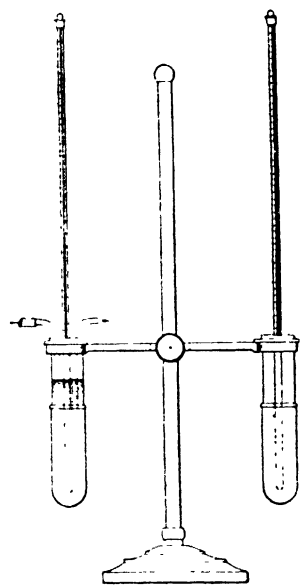


Fig. 22.—Dew-point Apparatus.

This form of apparatus is difficult to manipulate, and for that reason the wet- and dry-bulb hygrometer is used to determine humidity. This consists of two thermometers, one of which carries a piece of wicking around its bulb which is wet with distilled water. As water is evaporated from this wicking, the temperature is lowered and the thermometer reads lower than the dry-bulb thermometer. This action is not complete if the thermometer or air is at rest, and hence the U. S. Weather Bureau recommends whirling these thermometers.

Fig. 23 pictures this type of instrument. It is sometimes called a sling psychrometer.

W. H. Carrier discussed the matter of psychrometric formulae in the Transactions of the American Society of Mechanical Engineers (A.S.M.E.) for 1911 (page 1005), and showed that the temperature of the wet bulb is such that the lowering of the temperature of the air and superheated water vapor in the atmosphere to this point will

liberate sufficient heat to vaporize the necessary moisture from the wicking to saturate the air at the wet-bulb temperature. This means

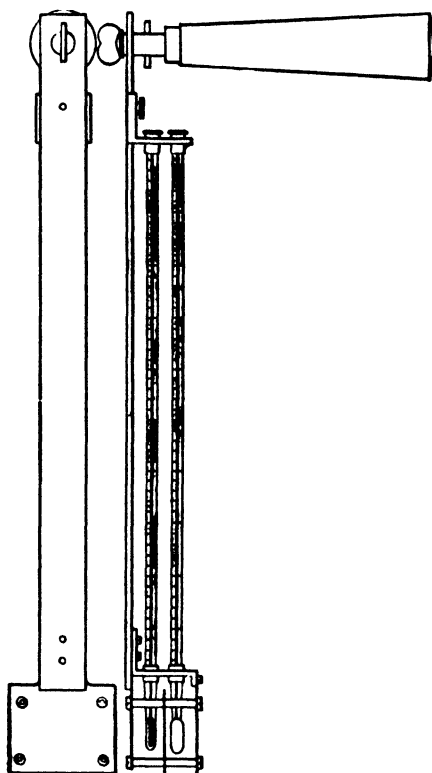


FIG. 23.—Sling Psychrometer.

that the wet-bulb temperature will be above the dew point. This wet-bulb temperature depends on the sum of the sensible and latent heat in the air and its vapor, and is independent of the relative proportions of them. Using this fact, Carrier gave the following for 1 lb of air, mixed with W pounds of water vapor:

Heat given up in cooling air and moisture

$$= (1 \times C_{pa} + C_{ps} \times W)(t - t') \quad (2-6)$$

Heat received by water at temperature of wet bulb

$$= h'_{fg}(W' - W) \quad (2-7)$$

C_{pa} = specific heat at constant pressure of air = 0.24112 + 0.000009 t ;

C_{ps} = specific heat at constant pressure of steam

$$= 0.4423 + 0.00018t \quad (\text{approximately});$$

t = temperature of air or dry bulb in degrees F;

t' = temperature of wet bulb in degrees F;

W' = weight of moisture per pound of air to saturate air at temperature t' ;

h'_{fg} = heat of vaporization of steam at temperature t' .

These may be equated giving

$$(C_{pa} + C_{ps}W)(t - t') = h'_{fg}(W' - W)$$

or

$$C_{pa}t + WC_{ps}(t - t') + Wh'_{fg} = C_{pa}t' + W'h'_{fg} \quad (2-8)$$

This equation indicates that the so-called "total heat" of the original mixture at dry-bulb temperature, consisting of the sum of the enthalpy of 1 lb of dry air above 0° F, the enthalpy to superheat the moisture associated with 1 lb of dry air from wet-bulb temperature to dry-bulb temperature and the enthalpy of vaporization at wet-bulb temperature of this moisture, is equal to the enthalpy of 1 lb of air above 0° F and the enthalpy of evaporation of its saturation moisture at the wet-bulb temperature regardless of the values of W . This means that the "total heat" in the air vapor mixture per pound of dry air contained remains the same for all mixtures having the same wet-bulb temperature. This fact is most important, and cooling on a constant wet-bulb line is often spoken of as adiabatic cooling. The heat for the evaporation of additional moisture comes from the mixture and not from the outside.

Solving Eq. (2-8) for W gives

$$W = \frac{h'_{fs}W' - C_{ps}(t - t')}{h'_{fs} + C_{ps}(t - t')} \quad (2-9)$$

This equation can be used to solve for W , the moisture content, when t and t' are known, as the other quantities can be found from tables of properties of steam and air.

It is to be noted that

$$W' = \frac{53.35 \times (t' + 459.6)}{144(\text{Bar} - p')} W'_t \quad (2-10)$$

W' = weight of saturation moisture per pound of dry air;

W'_t = weight of 1 cu ft of saturated steam at temperature t' ;

Bar = barometric pressure in pounds per square inch;

p' = vapor pressure at wet-bulb temperature in pounds per square inch;

$(\text{Bar} - p')$ = partial pressure on air in pounds per square inch.

If W' and W are known, the drop in temperature (wet-bulb depression) due to saturating the air at the wet-bulb temperature is given by

$$(t - t') = \frac{h'_{fs}(W' - W)}{C_{ps} + C_{ps}W} \quad (2-11)$$

The amount of water to be absorbed is given by

$$W' - W = \frac{(C_{ps} + C_{ps}W)(t - t')}{h'_{fs}} \quad (2-12)$$

Carrier, in addition to these, derived the theoretical formula for the partial vapor pressure, p :

$$p = p' - \frac{(\text{Bar} - p')(t - t')}{2800 - 1.3t} \quad (2-13)$$

It is to be noted that although p , p' and Bar are expressed in pounds per square inch, any units, such as inches of mercury, may be used in this expression.

On account of the error in the thermometer readings of the ordinary sling psychrometer, Carrier makes 1.6% correction, giving the equation for the partial pressure p from such readings.

$$p = p' - \frac{(\text{Bar} - p')(t - t')}{2755 - 1.28t} \quad (2-14)$$

As the moisture is superheated for low humidity it may be considered as a perfect gas and then the relative humidity will be the ratio of the actual partial vapor pressure to that of saturated air at dry-bulb temperature. If p_t is the saturation pressure at the dry-bulb temperature t , the relative humidity is

$$\rho = \frac{p}{p_t} = \frac{p'}{p_t} - \frac{\text{Bar} - p'}{p_t} \frac{t - t'}{2755 - 1.28t'} \quad (2-15)$$

From the equation it is seen that the effect of barometric pressure changes cannot be great and may be neglected in many problems.

The value of the hygrometer and the psychrometric formulae resulting lies in the information they give of the amount of water vapor contained in the air, and hence the condition of the air for use in a ventilating system or for other service. If, for instance, air partially saturated is brought in contact with finely divided water, evaporation will immediately take place, which will cool the air to the temperature of the wet bulb of the hygrometer. If, on the other hand, cold saturated air is warmed in a heater or by a steam coil, the moisture contained in the cold air is not sufficient to saturate the warm air, and the unpleasant sensation of excessively dry air is produced unless the air is humidified in some way.

Air which is very warm may be cooled in some processes and the capacity for water vapor may be much less than the actual quantity contained at the higher temperature and hence this reduction of temperature will cause precipitation. By finding the number of pounds of water vapor per cubic foot or per pound of dry air at a given tempera-

* In practice Carrier uses 2800 and 1.3 for the constants in (2-15).

ture from charts or tables, the value may be compared with the capacity at another temperature, and the relative humidity, quantity of vapor to be added or subtracted or any other property of the vapor at the new temperature may be found.

The amount of moisture in the air is an important item to consider in connection with the air for ventilating purposes. If the relative humidity is low, evaporation will take place from the surface of the body, producing dry skin or dry mucous membrane in throat or nose, and at the same time the temperature of the body will be lowered as a result of this evaporation. If, on the other hand, the air is saturated the body will feel damp and clammy. In both instances the air is objectionable. The usual amount of moisture to make the room comfortable should be such as to give a relative humidity between 30 and 60%.

In all these problems the volume of air will change, owing to temperature changes, so that the moisture per cubic foot would change, although there is no addition or abstraction of moisture; hence, it is far better to report the moisture as specific humidity, that is, as grains per pound of dry air, as this quantity can be changed only by adding or removing water vapor.

The pressure exerted by the water vapor at saturation is equal to the steam pressure at the temperature considered as given by any set of steam tables. This is usually known as vapor pressure or vapor tension. If the air is not saturated, the pressure is equal to that of the saturated vapor multiplied by the relative humidity, as shown in Eq. (2-15). Thus, if p_t = pressure of saturation and ρ = the relative humidity, the actual vapor pressure p is given by

$$p = \rho p_t \quad (2-16)$$

A similar formula may be used to find the weight in pounds of actual water vapor W_v in 1 cu ft of air, namely

$$W_v = \rho W_t \quad (2-17)$$

Where W_t is the weight of 1 cu ft of saturated steam at given temperature. Now the volume of 1 lb of dry air associated with water vapor is given by the equation

$$v_a = \frac{53.34(t + 459.6)}{144(\text{Bar} - p)} \quad (2-18)$$

where

v_a = volume of pound of dry air in cubic feet;

t = temperature of air in degrees F;

Bar = barometric pressure in pounds per square inch;

p = partial pressure of moisture in pound per square inch.

Hence the weight of moisture, W_{va} , associated with 1 lb of dry air is

$$W_{va} = \frac{53.34 \times (t + 459.6) W_t}{144(\text{Bar} - p)} \quad (2-19)$$

$$= \frac{53.34(t + 459.6)\rho W_t}{144(\text{Bar} - \rho p_t)} \quad (2-20)$$

For saturation

$$W_{va} = \frac{53.34(t + 459.6) W_t}{144(\text{Bar} - p_t)}$$

$$\frac{W_{va}}{W_{vt}} = \frac{\rho(\text{Bar} - p_t)}{(\text{Bar} - \rho p_t)} \quad (2-21)$$

$$\rho = \frac{W_{va}}{W_{vt}} \left(1 + \frac{1 - \rho}{\text{Bar} \cdot p_t - 1} \right) \quad (2-22)$$

The value of ρ might be 7% greater than the quantity W_{va}/W_{vt} at 100° because of the second term of the bracket.

W , being such a small quantity when expressed in pounds, may be reduced to grains by multiplying by 7000.

These formulae for the weight of moisture in the air in grains are:

$$W'' \text{ (grains per cu ft)} = 7000 \rho W_t = 7000 W_v \quad (2-23)$$

$$W' \text{ (grains per lb dry air)} = \frac{7000 \times 53.34(t + 459.6)\rho W_t^*}{144(\text{Bar} - \rho p_t)} \quad (2-24)$$

In cooling air and its moisture by some outside cooling means, the value of W_{va} does not change as the whole mixture is cooled so that the value of W_{va}/W_{vt} may be written as

$$\frac{W_{v \text{ dew}}}{W_{vt}}$$

* The 1936 Guide of the A.S.H.V.E. states that an examination of the Keenan steam table indicates that the weight of moisture in grains held by 1 cu ft of air may be expressed quite closely as

$$W''' = \frac{5785 p'_s}{t + 460}$$

where p'_s = partial pressure at t° F in inches of mercury and that the grains of moisture with 1 lb of dry air is given by

$$W' = 4354 \left(\frac{p'_s}{\text{Bar} - p'_s} \right)$$

These equations are much simpler than (2-23) and (2-24).

This would give

$$\rho = \frac{W_{v \text{ dew}}}{W_{vt}} \left(1 + \frac{1 - \rho}{\text{Bar}/p_t - 1} \right) \quad (2-25)$$

Since there is some change in volume as the air is cooled to the dew point the relative humidity is given approximately by

$$\rho = \frac{W_{\text{dew}}}{W_{vt}} \times \frac{T_{\text{dew}}}{T_a} \quad (2-26)$$

$$T = t + 459.6$$

If the relative weights of air and moisture do not change, the partial pressures of dry air and of the moisture remain the same by Dalton's law so that $p = p_{\text{dew}}$ and

$$\rho = \frac{p_{\text{dew}}}{p_t} \quad (2-27)$$

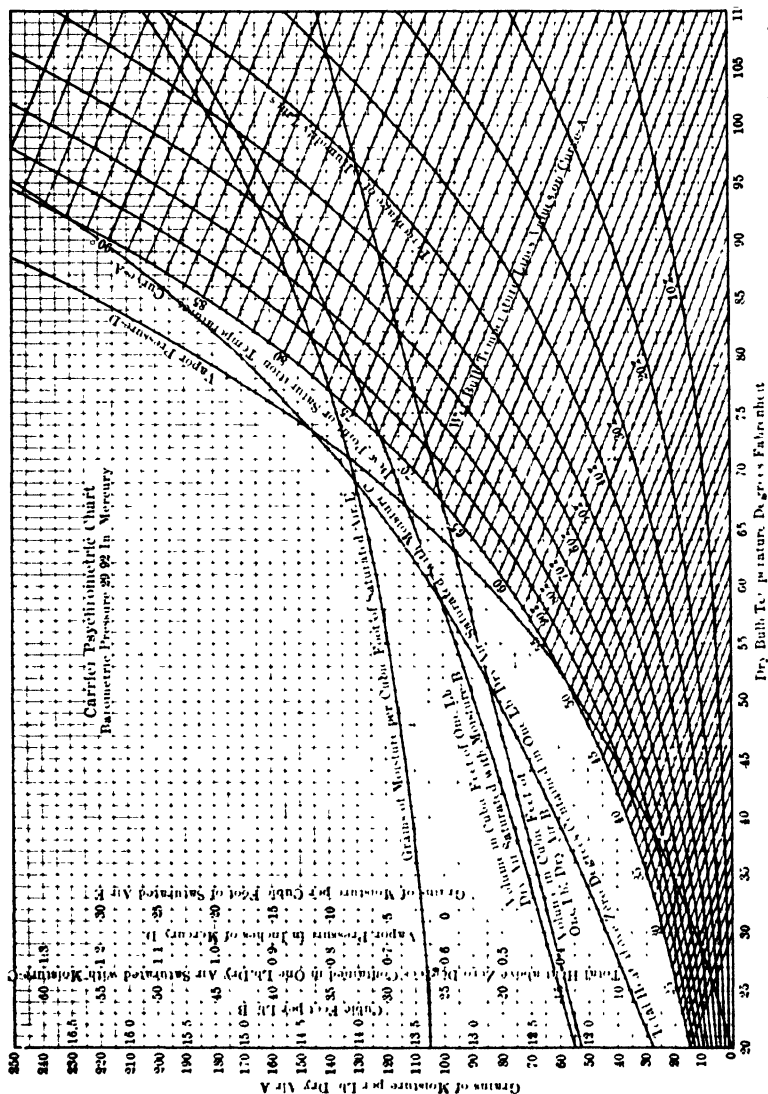
The quantity called total heat per pound of dry air and its associated moisture at the wet-bulb point

$$C_{pa}t' + W'h'_{fg}$$

is given by Carrier as the total heat per pound of dry air at the temperature t' , and it must be remembered in working problems where the air and its associated moisture are cooled to a point below the wet-bulb temperature that the change of enthalpy of the condensed liquid must be allowed for in the computations.

For many years the set of curves presented by Carrier in this paper (1911) have been used by engineers. This is called the Carrier Psychrometric Chart and is shown in Fig. 24. This figure is drawn for a barometric pressure of 29.92 in. of mercury, and its abscissae represent dry-bulb temperature readings, that is, the temperature of the air and its contained moisture; the ordinates of the diagram have different values depending on the curves considered. In the first place, the curves running upward from left to right and marked by percentages represent relative humidities. The amount of water vapor in grains associated with 1 lb of dry air (specific humidity), at various percentages of relative humidity, are found by going horizontally from a given point to the left hand by the first scale on the left. The point at which the horizontal line crosses the 100% line will give the temperature of the dew point for a given moisture content per pound of dry air. Points on the upper curved line of 100% relative humidity will give the weight of water vapor in grains per pound of dry air to saturate the air at these different dry-bulb temperatures. Of course

under this condition the air is at the dew point and also the wet bulb will have the same temperature as the dry bulb. From this curve *A* and extending downward to the right a series of lines are marked wet-



Courtesy of Carrier Engineering Corporation

Fig. 24.—Carrier Psychrometric Chart.

bulb temperatures. These curves show that, as the dry-bulb temperature associated with a given wet-bulb temperature rises, the relative humidity becomes smaller, or with a given dry-bulb temperature higher relative humidities are associated with higher wet-bulb temperatures.

The difference between the temperature readings of the dry- and wet-bulb thermometers is called the depression of the wet bulb, and it is seen that the less the relative humidity the greater the depression of the wet bulb. It must also be remembered from what has been shown earlier that the heat or enthalpy of 1 lb of air above 0° F, plus the heat of vaporization at the wet-bulb temperature of the amount of moisture per pound of dry air, is the same at any condition along one of the wet-bulb lines; and consequently above the saturation end of this wet-bulb line, the line *C* is laid off, which gives at the left the total heat of the mixture in Btu per pound of dry air in the mixture (enthalpy of dry air above 0° plus the enthalpy of vaporization of the moisture contained in 1 lb of dry air saturated with moisture). The values of the ordinates of this curve *C* are indicated in the third set marked at the left in Btu associated with pound of dry air.

There are two curves marked *B*, one which gives the volume of 1 lb of dry air in cubic feet, and one for the volume of 1 lb of dry air saturated with moisture. These curves are read from the dry-bulb temperature lines for the given temperature. The difference between the ordinates of the two *B* curves at any given temperature indicates the increase in volume due to the moisture added to take its part of the atmospheric pressure. This difference is multiplied by the decimal value of the relative humidity to determine approximately the quantity to add to the dry volume so as to obtain the volume of 1 lb of dry air and its associated vapor when the air is not saturated.*

* The exact formula for volume is given as

$$\begin{aligned}
 v_p &= \frac{v_s v_d}{v_s - \rho(v_s - v_d)} \\
 v_d &= \frac{RT}{p_b} \quad \text{or} \quad p_b = \frac{RT}{v_d} \\
 v_s &= \frac{RT}{p_b - e} \quad \text{or} \quad p_b - e = \frac{RT}{v_s} \\
 v_p &= \frac{RT}{p_b - \rho e} \quad \text{or} \quad p_b - \rho e = \frac{RT}{v_p} \\
 \rho e &= \rho RT \left[\frac{1}{v_d} - \frac{1}{v_s} \right] \\
 v_p &= \frac{RT}{RT \left[\frac{1}{v_d} - \rho \left(\frac{1}{v_d} - \frac{1}{v_s} \right) \right]} = \frac{v_s v_d}{v_s - \rho(v_s - v_d)}
 \end{aligned}$$

By division this becomes

$$\begin{aligned}
 v_p &= v_d + \rho \frac{v_d}{v_s} (v_s - v_d) + \rho \frac{v_d}{v_s^2} (v_s - v_d)^2 + \dots \\
 &= v_d + \rho(v_s - v_d) \quad (\text{approximately})
 \end{aligned}$$

The curve *D* gives the vapor pressure from the moisture saturating the air at a given dry-bulb temperature reading, and its scale is the fourth one on the left. At any partial saturation the vapor pressure may be taken as that shown by curve *D* multiplied by the decimal relative humidity.

The curve *E* gives the grains of moisture per cubic foot to saturate the air. This quantity may be needed in certain problems although it is not as valuable as the values from the *A* curve or its fractional parts, i.e., grains per pound of dry air.

To use the curves, suppose that the dry-bulb reading is 70° F and the wet-bulb reading 55° F; the curves then show that the relative humidity is 37%, the moisture amounts to 40 grains per pound of dry air, found by carrying the horizontal line at the crossing of 70° and 55° to the left. The dew point, found where the horizontal cuts the saturation line, is 42.2° F.

The heat per pound of dry air and the associated moisture is found by following up the wet-bulb reading until it meets the saturation line and then carrying a vertical from this point to the line *C*. In this case the reading at the left at the point at which the 55° line meets *C* is 22.9 Btu per lb of dry air. Remember that the heat of the liquid is not included in this value.

The volume of dry air at 70° F is 13.35 cu ft per lb, and that of the air saturated with moisture is 13.69 cu ft. The difference is 0.34 cu ft, and for 37% relative humidity the volume of the air is $13.35 + 37\%$ of 0.34 = 13.48 cu ft per lb of dry air.

The vapor pressure for saturated air at 70° F is 0.738 in. of mercury from *D* curve, and at 37% relative humidity the vapor pressure would be $0.738 \times 0.37 = 0.273$ in. of mercury. This could also be found by reading the pressure on *D* directly over the dew point at 42.2° F. This gives 0.273.

The vapor to saturate the air at 70° from *E* curve is 8 grains per cu ft. At 37% relative humidity the approximate amount is $8 \times 0.37 = 2.96$ grains per cu ft.

From 40 grains per lb of dry air and 13.48 cu ft per lb of dry air, the amount of moisture per cubic foot is $\frac{40}{13.48} = 2.965$. These two methods check.

From such quantities as have been found from the curves above, from given data or from the converse of these problems, data needed for heat demand, volumes and water supply may be found.

Another chart, used in the same manner as described above, is the Bulkeley Psychrometric Chart, Fig. 25, presented by Mr. C. A. Bulkeley to the A.S.H.V.E. in 1926. It is a modified form of the Carrier Chart in which Curve *A*, or the saturation line, has been made a straight line on a semilog or aritholog paper in which the vapor pressure in millimeters of mercury is laid off logarithmically on a vertical line from 0.9 mm at the bottom left to 600 mm at the top. The horizontal length is made any desirable amount. A straight line is drawn from 0.9 mm at the left to 600 mm at the right, and on it are marked points

of temperature from 10° to 200°, placed at the points of proper vapor pressures. These points fix an irregular but correct temperature scale for dry-bulb temperatures at 10° intervals on the base line by drawing vertical lines from the points to the horizontal axis. The lines of unit

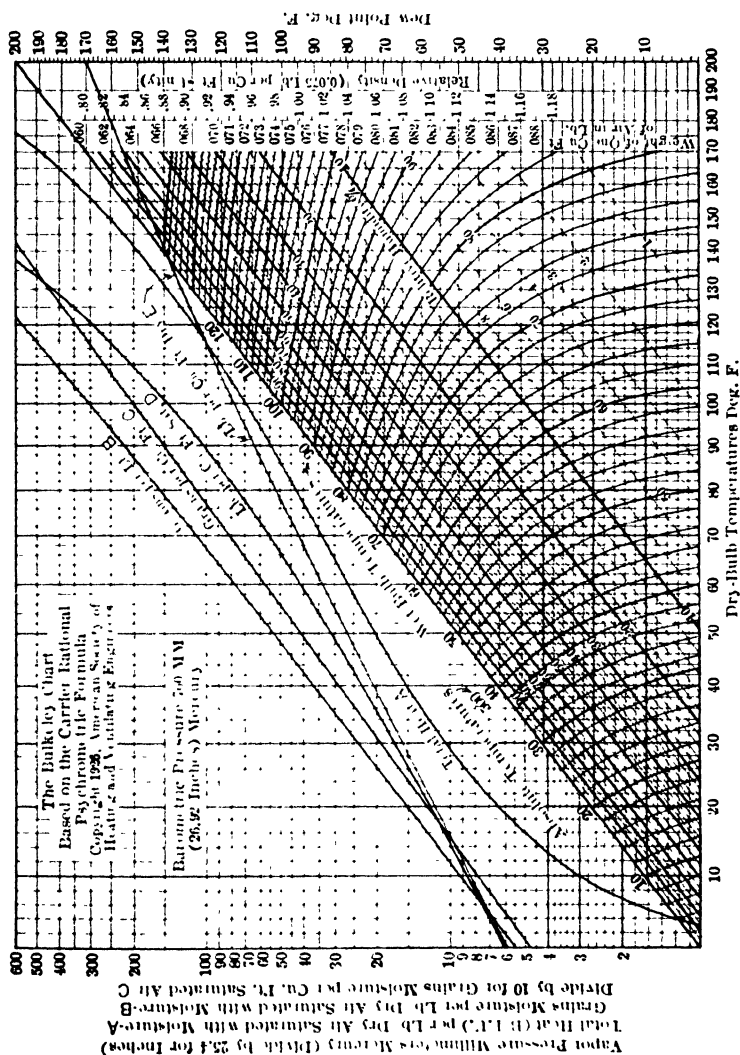


FIG. 25.—Bulkeley Psychrometric Chart. Copyright, 1926, American Society of Heating and Ventilating Engineers—Reprinted by Permission.

temperatures are found by dividing these intervals into ten equal parts. The relative humidity lines now become parallel to the straight saturation curve drawn, since, for the vertical logarithmic scale, $\log (\text{Relative humidity} \times \text{Pressure}) = \log \text{Relative humidity} + \log \text{Pressure}$, and the

line of fixed relative humidity differs from the saturation line by a constant quantity, log Relative humidity, at any moisture content. Starting on the saturation line at some temperature, as the wet-bulb temperature with 100% relative humidity, the relative humidity at a number of other dry-bulb temperatures is computed for this wet-bulb temperature, and these points of dry-bulb temperatures and relative humidities are marked with this wet-bulb temperature. Joining these points the wet-bulb-temperature lines become curved lines in place of straight lines of wet-bulb temperature of Carrier.

In addition to these charts, Table A-2 * gives various properties of air and air mixtures. Tables A-3-1, A-3-2 and A-3-3 give the properties of steam. Table A-2, modified from a table, "Mixtures of Air and Saturated Water Vapor," in Goodenough's "Tables of Steam and Ammonia," and Tables A-3-1 to A-3-3, modified from Keenan's Steam Tables, are presented for those who prefer to work with tables. The atmospheric pressure used is 29.92 in. of mercury, and the effects of other barometric or space pressures should be examined in problems involving air under different pressures to make certain that appreciable changes do not occur. The enthalpy of the mixtures given in Table A-2 includes the enthalpy of the liquid for the water vapor.

EFFECTS OF AIR CONDITIONS

Experiments have been made at the research laboratory of the A.S.H.V.E., associated with the laboratory of the U. S. Bureau of Mines in Pittsburgh, to ascertain the air conditions in a room to give satisfaction to the occupant and maintain a healthful and stimulating environment. For many years such experiments as those made by the New York Commission on Ventilation have been tried throughout the world to determine the elements of good ventilation; they have been found to be temperature, humidity, air movement, cleanness, freedom from toxic and other objectionable gases or fumes and from odors and possibly the change in the ionic content of the air.

Carbon dioxide, as shown earlier, is merely an indication of the amount of air supply per person, and experiments have shown that the emissions from the occupants do not have toxic elements. Body odors may be present when the fresh air supply is small, and then the air is objectionable and will lead to loss of efficiency and discomfort of those who breathe it. Of course, dangerous bacteria may be given off by infected persons in the space being heated, but in general these are not carried by the air.

* The tables appearing in the Appendix are marked A.

Temperature, movement of air and humidity are most important factors, as the heat produced by the occupants of a room must be removed by the surrounding air. The normal temperature of the body is 98.6° F, and the heat produced by the vital actions going on within the body must be removed by radiation, conduction, convection and evaporation of perspiration, as in general the surrounding air and bodies are at a lower temperature. It is quite evident that, when air of high humidity is moved over the body at a low speed, perspiration will be absorbed at a very low rate, if at all. Then, if the velocity of air movement be increased, or if the relative humidity be decreased but the original velocity maintained, the rate of absorption of this perspiration will be increased, and because of the heat needed for the evaporation of the perspiration, the heat removed from the body is increased and the subject therefore feels cooler.

The temperature of the air at this time must also be considered as it fixes the maximum possible content of vapor as well as the rate at which the heat may be removed by transfer to the air as sensible heat. This rate of heat transfer depends on the thickness of the film next to the clothing and skin, and this in turn is fixed by the velocity of air movement; the higher the velocity the greater the rate of heat transfer. With high velocities, drafts are experienced by the occupants of the room, and reflex actions take place which may lead to discomfort as bodily temperatures are changed. The same may be said of the temperatures of the ambient air. Reflex actions which take place in the occupants control the temperature of the skin at different parts of the body and that of the nasal and oral passages as well as the action of the circulatory system and sweat glands at the surface of the body. The A.S.H.V.E. laboratories have determined the increases in the pulse rate, in the temperature of the rectum and in loss of weight of persons moved from rooms at 70° F into rooms at higher temperatures in which they remained at rest or at work. These increases of temperature limit the activity of the body which was shown by the reactions of the subjects. The New York Commission on Ventilation has shown that when a person was introduced suddenly into a region of lower temperature there was lowering of the average skin temperature, that of the hands and feet falling much below the average. Also, the mucous membrane of the nose dropped in temperature, but that of the mouth did not change much. It found that localized draft on head or feet caused little change in temperature of trunk, thighs and legs; chilling of the head alone lowered the temperature of the mucous membrane of the nose and mouth but these increased when the feet were chilled.

In making these experiments on effects of temperature, curves of effect versus wet-bulb temperature, dry-bulb temperature or other bases were very irregular. In 1923, however, F. C. Houghten and C. P. Yagloglou of the A.S.H.V.E. laboratory determined the equality, as far as comfort was concerned, between temperature of saturated still air and air at other temperatures and relative humidities. Curves of equal comfort were called lines of effective temperature (*ET*). This determination was made by maintaining air of practically no velocity at a given temperature in one psychrometric chamber with a certain relative humidity or wet-bulb temperature and then varying the conditions of dry-bulb temperature but with a fixed relative humidity of still air in another psychrometric chamber of form similar to that of the first one, until the subjects, used in the experiment to visit alternately one chamber and the other, reported that there was the same temperature in each. When lines joining these points were carried out to the saturation line the temperature at this point gave the value of the effective temperature of this line. Thus were obtained effective temperature curves. On plotting the effects of heat and other phenomena versus effective temperatures, the effects took on a real meaning and some interpretation of these experiments could be made.

Houghten and Yagloglou reported in 1924 on the cooling effect of velocity; in 1925 C. P. Yaglou and W. E. Miller reported on the effective temperature with clothing, and in 1926 Houghten, Teague and Miller reported on effective temperature for persons at work when lightly clothed. With velocity of appreciable amount the effective temperature line determined with reference to still air was moved to a higher point on the saturation line and such lines were called lines of equivalent effective temperature. Finally, these lines were called effective temperature lines at a given velocity. Effective temperature corresponding to a given condition of air temperature (dry bulb), relative humidity (wet and dry bulb) and air movement is the temperature of still air (velocity of 15 to 25 fpm) saturated with water vapor which gives the sensation of the same degree of warmth to persons clothed in customary apparel.

Yaglou and Miller have prepared a diagram, Fig. 26, by which the effective temperature may be found from wet- and dry-bulb thermometer readings at different velocities of air movement for individuals of the United States normally clothed for indoor sedentary or light muscular work when subject to heating by warm air, direct steam and hot water radiation, or plenum systems. The effective temperature is found at the intersection of the straight line joining the points of wet- and dry-bulb temperatures with the line of velocity of the air.

The values of these effective temperatures are the temperatures to be used in studying the effects of ventilation on occupants of spaces.

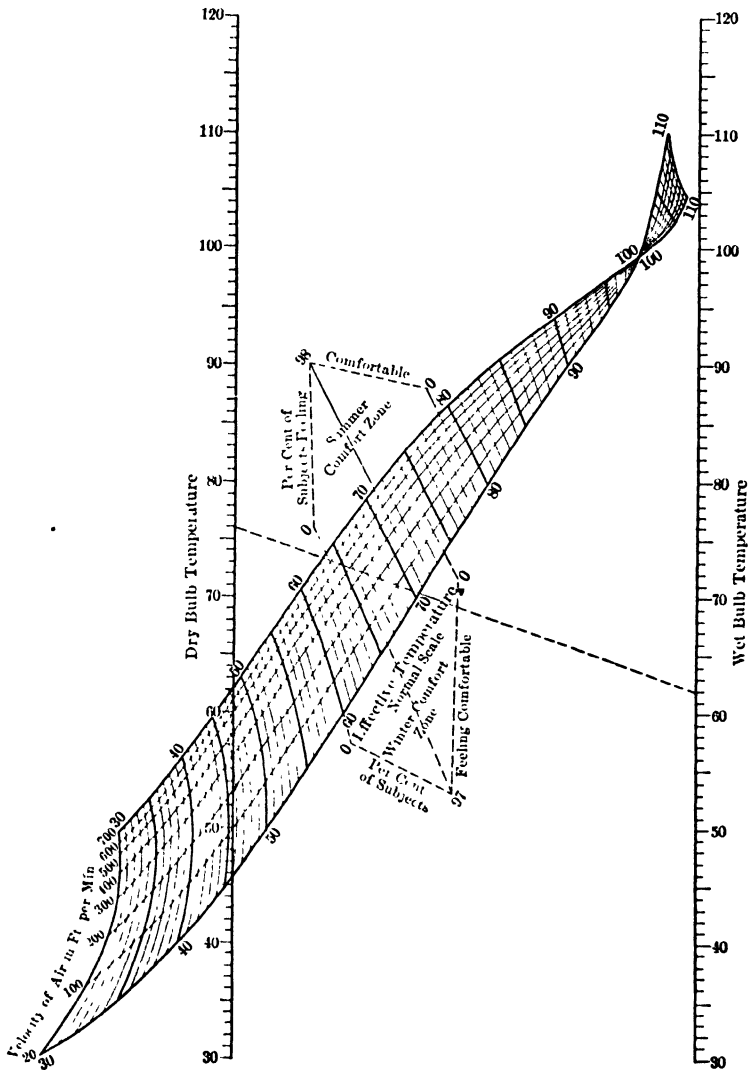


FIG. 26.—Effective Temperature Chart, Applicable to Inhabitants of the United States, with Customary Indoor Clothing and at Sedentary or Light Muscular Work. Copyright, American Society of Heating Engineers. See A.S.H.V.E. Trans. Vol. 31, 1925 and 38, 1932.

Experiments have been made at the A.S.H.V.E. laboratory and the Harvard School of Public Health to determine the comfort zone for

summer and winter in which a large number of persons were examined. The results of these tests are given on Fig. 26. The results for summer and winter are different because of the effect of seasonal atmospheric conditions on the subjects as well as the types of clothing. The center line of 71°F represents the effective temperature at which 98% of the persons reported comfort in summer and that of 66° at which 97% reported comfort in winter with air movement of 15 to 25 fpm (still air). The limiting lines at which none reported comfort were 64°F and 79°F in summer, and 60°F and 74°F in winter. The comfort zone within which at least 50% of the persons were comfortable extends from 66° to 75°F in summer and from 63° to 71°F in winter. The air movement, as stated above, was quite low, 15 to 25 fpm, which is almost still air. The relative humidities should be between 30% and 70%. Some think that 65% relative humidity should be used to maintain a healthful condition of the atmosphere. The ranges of these zones have been marked on the chart of effective temperature, Fig. 26, for use.

In applying these comfort-zone figures it must be realized that it takes some time for the human body to reach a new condition of effective temperature. In applying the comfort-zone temperatures, regard should be paid to the change in effective temperature and to the time during which the occupant will be in the air-conditioned atmosphere. For homes, offices or schools the comfort zone may be used, but with short occupancy as for stores, restaurants, theatres and public rooms of offices, the upper 50% limit, 75°ET , of the comfort zone should be used where outdoor temperatures of 90° and above are found, while in the eighties 71°ET could be employed.

Houghten and his associates have found that when people are seated at rest the forehead will feel clammy at 73°ET with moist air, and at 75°ET (87°F dry bulb) with comparatively dry air (20% relative humidity). When people are at work these effective temperatures become 59.0°F and 69.5°F , respectively.

The temperature, t_s , of the space to be heated should be governed by the foregoing principles only when the occupants are to be considered, but in many cases the specific use of the space fixes the temperature and humidity which should be maintained to obtain a certain product and secure production and continuity of service. These requirements for various industries are given in Table 2 of Chapter 40 of the 1936 Guide of the A.S.H.V.E., to which reference is made. Table 1 of that chapter gives the moisture content of various materials in percentage of weight when these materials are exposed to different humidities.

As a guide to the designer the usual temperatures of spaces for definite uses are given:

TABLE 2-1

DRY-BULB TEMPERATURES OF ROOMS

Warm air baths.....	122° F	Laboratories, light work ..	70° F
Steam baths.....	113° F	Toilets.....	68° F
Operating rooms (surgical)....	85° F	Restaurants.....	68° F
Examination rooms (medical).	85° F	Auditoriums ..	66° F
Greenhouses.....	77° F	Kitchens. .	66° F
Massage rooms ...	77° F	Ballrooms	65° F
Swimming pools ..	75° F	Shops, light work..	64° F
Bath rooms.....	72° F	Churches... ..	64° F
Hospital rooms ...	72° F	Sleeping rooms..	60° F
Houses, offices ...	70° F	Entrances, corridors.....	60° F
Classrooms, schools ..	70° F	Laboratories, machinery	55° F
Sewing rooms.....	70° F	Gymnasiums, work shops	55° F

When occupants are likely to be heavily clothed, as in churches, when muscular work is being done or when machinery in laboratories gives out heat which is not considered in the computations, a low design temperature may be used as indicated above.

In addition to the above, experiments have been made by C. P. Yaglou and others on the ionic content of occupied rooms showing that this content is reduced rapidly to a low level as soon as the room is occupied, and the content is restored soon after the room is unoccupied. The effect of this on human comfort has not been determined although from present evidence ionic content does not seem to be associated with comfort. Loeb states that the physiological effects of various ions as they influence air conditioning are absolutely negligible because of the amount of foreign matter which must be carried into the lungs to introduce appreciable changes. He believes that more study should be given to the effects on the occupants of rooms of small quantities of ozone, N_2O_4 , SO_2 , NH_3 , CO_2 , CO and water vapor in the air.*

The cost of heating or cooling fresh air in quantities necessary for proper distribution and maintenance of uniformity of composition without any recirculated air is very great in winter or summer, so that partial recirculation has been tried in a number of installations, and should be used where possible. Recirculated air must be dehumidified whenever the humidity is increased by occupancy or use, and if body odors persist even after washing, recirculation may not be possible. The A.S.H.V.E. Code requires at all times that fresh outside air amounting to 10 cfm per person must be added to the recirculated air. In

* See Journal Section, Heating, Piping and Air Conditioning, Oct., 1934, p. 437.

some tests subjects have reported that recirculated or partially recirculated air lacks tone and feels stale, although others have reported it better than or as good as entirely fresh air. It is to be remembered that even the fresh outside air is taken through washers and over metal heating devices and through metal ducts. To remedy the lack of tone some have suggested treatments with ultra-violet light, but there is no evidence of the value of such treatment. Small amounts of ozone have been used to kill odor and lend tone to the air. Instances are quoted in

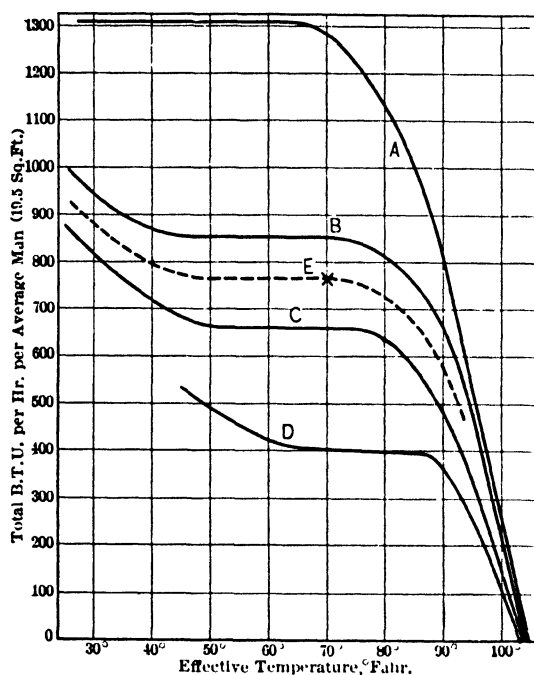


FIG. 27.—Relation Between Total Heat Loss per Hour from Human Body and Effective Temperature for Still Air. Copyright, American Society of Heating and Ventilating Engineers. See A.S.H.V.E. Trans., Vol. 37-1931.

ratus. To give some idea of the heat released by the average man as a function of effective temperature, Houghten, Teague, Miller and Yant in the A.S.H.V.E. Transactions for 1931 gave the following curves, Fig. 27: A for men doing 66,160 ft-lb of work per hr; B for men doing 33,075 ft-lb of work per hr; C for men doing 16,538 ft-lb of work per hr, and D for men at rest. This shows that effective temper-

the recent Transactions of the A.S.H.-V.E. on the beneficial effect of ozone, and a few years ago the use of ozone in movie houses was frequently reported. Loeb * has suggested that ozone may be the cause of serious respiratory afflictions since, when sufficient quantities are used as a germicide, they may be destructive of human tissues.

Heat from occupants must be considered in recirculated air either as a part of the heat supply for losses from walls or part of the heat to be removed in the air-conditioning apparatus.

* *Ibid.*

ature influences the heat released, which gradually decreased as the effective temperature increased.

The heat produced is largely removed by convection and radiation, but as the temperature rises perspiration and its evaporation also remove much of the heat as shown in Fig. 28 of Houghten, Teague, Miller and Yant. This figure not only indicates the latent heat per hour from the evaporation of perspiration of the average man but also the grains of moisture per hour added by him to the space occupied. This latent heat, although not present as sensible heat, is reflected in the greater heat per pound of dry air and its associated moisture used to determine the

amount of refrigeration needed in cooling, although for effect on room temperature the sensible heat from persons is used.

These tests were originally computed on a 1 sq ft surface basis as the areas varied from 10 sq ft for a person $4\frac{1}{2}$ ft tall, weighing 45 lb, to 26 sq ft for a $6\frac{1}{2}$ ft person weighing 235 lb. The average results are given for 19.5 sq ft, which hold for a person 68 in. in height, weighing 150 lb.

Both heat and moisture production must be considered in the problems involving large numbers of persons such as are found in churches, schools, auditoriums and stores.

If lights or machines are operating in any space considered for air conditioning, the heat produced by these must be taken into account.

All the electrical energy used in electric lights is changed into heat, and machines using electric motors usually change the output of the motor into frictional heat. In these cases, 1 watthour produces 3.41 Btu, which must be allowed for in treating the air.

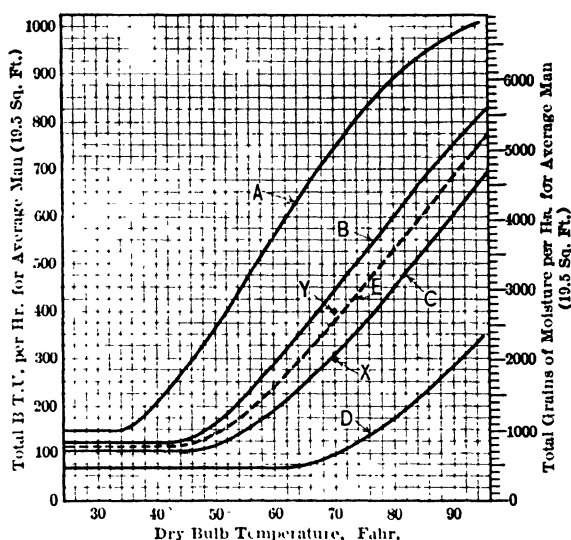


FIG. 28.—Latent Heat and Moisture Loss per Hour from Human Body by Evaporation in Relation to Dry Bulb Temperatures. Copyright, American Society of Heating and Ventilating Engineers. See A.S.H.V.E. Trans., Vol. 37-1931.

Earlier values of average heat produced by persons are given below:

TABLE 2-2
HEAT RELEASED FROM PERSONS (Total)

Adults asleep	260 Btu per hr
Adults seated	380 " " "
Adults at work	670 " " "
Adults at hard work	1500 " " "
Adults, in old age	360 " " "
Children	240 " " "
Infants	63 " " "
Walking 3 mph *	1049 " " "
Walking 5 mph *	2530 " " "
Slow run *	2285 " " "
Maximum exertion *	3333 to 4762 "

* Given by Houghten as metabolic rate, which includes the external work which may not be included in the space to be heated. The external work amounts to about 11% of the metabolic rate. The value of external work may be used to spot point *x* on curve *E*, Figs. 27 and 28, for the construction of *E*.

Usually where gas, oil or steam is employed for industrial heat supply or electrical energy for power or light very little of the energy of these sources is used to increase the energy of the materials produced, so the full heat values of these must be considered in conditioning air of a space in which they are employed. The average heats per pound or cubic foot are given below:

TABLE 2-3
HEAT VALUES

Anthracite coal	12,500 Btu per lb
Bituminous coal	14,500 " " "
Fuel oil	18,750 " " "
Steam	1,000 " " "
Natural gas	1,000 " " cu ft
Illuminating gas	575 " " " "
Producer gas	150 " " " "
One hp-hr	2,543 Btu
One kw-hr	3,410 Btu
One Welsbach burner uses 3 cu ft gas per hr	
One fish-tail burner uses 5 cu ft gas per hr	

Where possible, it will pay in all ventilating systems to arrange for recirculation to the maximum amount (all but 10 cfm per person), but provisions should be made to regulate the quantity of fresh air, as satis-

factory air is the aim of all ventilating systems, and if this means no recirculation the system must be operated in this expensive way.

Air free of dirt and gaseous impurities is one of the important features of ventilation. Most solid suspended matter in air used for ventilation may be in amounts from about 0.3 grain per 1000 cu ft in rural districts to 0.75 grain per 1000 cu ft in cities and twice this amount near industrial plants. These materials are usually small, and at times the size is such that a filter of cloth will care for the particles.

The amount of dust per cubic foot is difficult to measure. One method is to draw a known amount of air through a small hole beyond which is a glass plate covered with a sticky material. The dust particles are then counted with the aid of a microscope, and the number per cubic foot is calculated. Dust may be collected on filter paper, in water or in porous crucibles, and weighed. There is no standard method. Fig. 29 gives a section of the Hill Dust Counter with its microscope and inhaling cylinder of known volume.*

The bacterial quality of the air is judged by allowing the air to come in contact with a culture medium for two minutes and then counting the colonies developed after a period of incubation at a standard temperature. For pathogenic germs the more extensive methods of the bacteriologist must be employed.

A simple form of filter for air purification consists of screens made of fine-mesh fabric of such extended area that the velocity of the air currents through it will be from 10 to 50 feet per minute (fpm). The fabric is hung in folds of such a large area that the velocity through it is many times less than the velocity in the duct leading to the filter.

* The U. S. Public Health Service uses the Greenburg-Smith Impinger in its study of dust content. In this, air is drawn into a tube of water, and the amount of dust collected from 10 to 30 cu ft of air is measured.

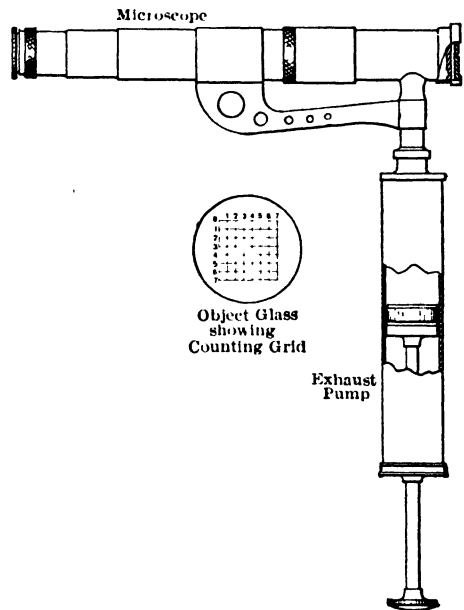
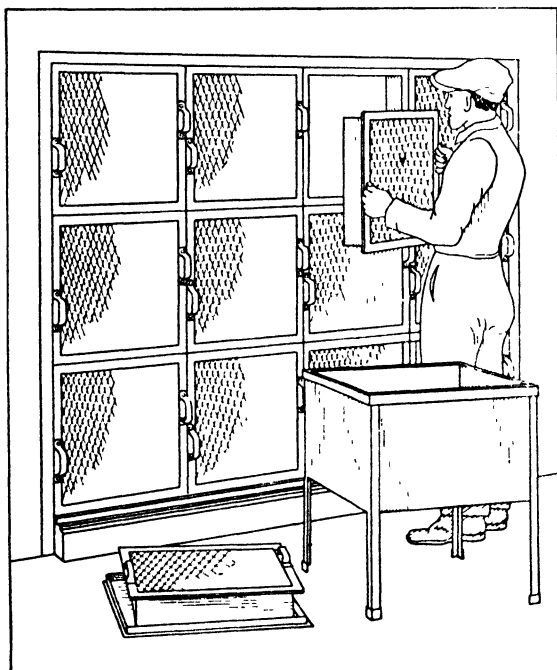


FIG. 29.—Hill Dust Counter.

This principle is used with paper filters where the paper is attached to a frame and the air passes through the sheets into a narrow passage formed between the sheets held in a supporting casing. The dry filters must be cleaned at times by rapping or by means of vacuum cleaners, although dirty paper filters may be replaced at little cost.

To remove this dust more effectively viscous filters have been constructed, composed of metal frames several inches deep holding some form of metal shavings and saturated with a viscous, non-volatile, germicidal and non-inflammable fluid which causes the dust particles



Courtesy of American Air Filter Co.

FIG. 30.—Viscous Filter Units.

to adhere to the shavings as the air is passed through the interstices between the metal chips. These filters, Fig. 30, made up in unit boxes, are placed in a supporting framework of such size that the average velocity over the complete frontal area is 200 to 400 fpm. The unit frames of interchangeable construction are of such size and weight that they may be removed readily by hand for cleaning at frequent intervals. This cleaning is done in a space near the filter by warm water of the necessary

chemical content for the removal of the dirt. The units are then treated with the viscous material before replacement. As the cells become dirty the resistance to air flow varies from $\frac{1}{3}$ to $\frac{2}{3}$ of an inch of water pressure, the original pressure drop for clean filters being from $\frac{1}{15}$ inch to $\frac{1}{8}$ inch of water. By gage readings of the pressure drop the operator may know when cleaning is necessary. In large units hand cleaning is continuous, the first unit frame being sufficiently dirty when the last has just been cleaned.

Viscous automatic filters are arranged with units forming a con-

tinuous curtain which rotates slowly and is carried through a trough of viscous fluid which serves to force out the dirt and recharge the units. Fig. 31 shows a section through one type of automatic filter in which cleaning action occurs at intervals determined by pressure drop.

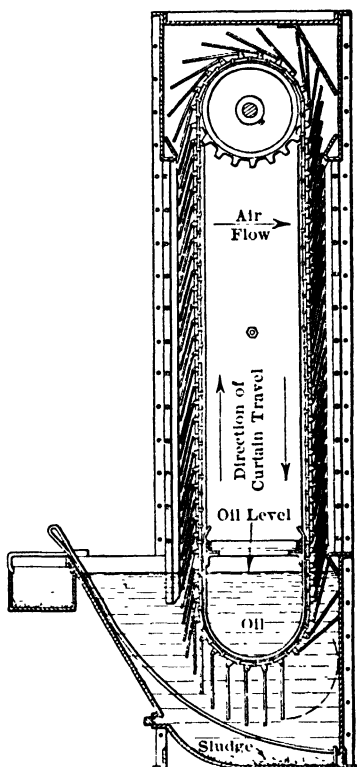
The framework supporting the filter is built of rigid members with work platforms and ladders or stairs attached for the removal of sections on large units. Each side of the filter should be in a chamber formed with access doors for inspection and maintenance and provided with necessary electric illumination.

The air washer is not so effective in removing dust as the modern filter and it does not act as a deodorizer. Its present use is primarily for control of humidity but it does contribute to the removal of some of the dirt and soluble gases.

When the washer is used as a scrubber it is customary to use several eliminators, such as Fig. 18, in series in connection with the washer. By adding two eliminators in front of the eliminator of Fig. 18, and by having the full area of these flooded with water, the three impingements on the bent plates eliminate the dust from the air and may be used as regainers of valuable material in the dust. The full pressure drop in the washer alone is given in inches of water as $h = (\text{fpm})^2 / 1,000,000$.

A fan gives the necessary pressure and velocity for overcoming the resistances in the filter, the washer, scrubber and eliminator, in the heating and tempering coils, the duct system and its associated parts. The recirculated air is admitted to the part of the suction system desired by the designer. In any case the designer fixes the work done by this washer as a cleaner and a controller of the moisture content of the air handled by it, so that a required condition in the occupied space is obtained.

The quantity of air circulated, as stated earlier, depends on the desired properties of the entering air and the maintenance of uniformity



Courtesy of Independent Air Filter Co

FIG. 31.—Double Duty Automatic Filter.

throughout the spaces which receive the air, and it must be computed for any particular installation. As a guide, however, the following values may be given:

TABLE 2-4
AIR CIRCULATED PER HOUR

For Adults per Person	
Hospitals. . . .	2400 cu ft
Auditoriums. . . .	1800 " "
Work shops	2000 " "
Waiting rooms. . . .	1000 " "
For Children per Person	
Hospitals.	1500 cu ft
Schools (by law). . . .	1800 " "
Auditoriums. . . .	1500 " "

By A.S.H.V.E. Code at all times:

Fresh air per person (at least) 600 cu ft

When occupants are not known:

Residences. . . .	1 to 2 changes per hour
Stores.	2 changes per hour
Offices.	2 changes per hour
Schools.	2 changes per hour
Auditoriums. . . .	2 changes per hour
Hospitals.	2 changes per hour
Kitchens.	4 changes per hour
Toilets.	2 changes per hour

The quality of air in a given room has been judged by finding the values of certain air qualities on arbitrary scales and then adding these, as has been suggested by Dr. E. V. Hill in his Synthetic Air Chart.*

Spaces so estimated have a numerical value for their qualities as each has been computed on the same basis, but the values given to the different properties are not of necessity the correct values. It is not evident that a variation of 5° F from some ideal temperature is as deleterious as 125,000 dust particles per cubic foot, or 130 bacteria colonies or 42 parts of CO₂ per 10,000. This means that the results obtained from different spaces are not truly comparable, but by this particular method numbers do represent values with the Hill allowances.

* A.S.H.V.E. Trans., vols. 23 and 26.

PROBLEMS

1. A room of 9600 cu ft contains 20 persons at clerical work. The CO_2 content of the room is 12 parts per 10,000. Find the amount of fresh air brought into the room per hour. Is this sufficient under the A.S.H.V.E. Code? How much air would you circulate for comfort? With the Wolpert air tester, 45 cc of outside air was required to change the color. How many cubic centimeters would be shown with the room air?

2. The outside air is at 80°F dry bulb and 65°F wet bulb. Find the dew point; the relative humidity; absolute humidity; specific humidity; total heat per pound of dry air and its moisture; enthalpy per pound of dry air and its moisture; specific volume. Find the vapor pressure.

3. Find the specific volume per pound of dry air, its moisture content and the enthalpy if delivered into a room at 60°F and 100% relative humidity. This air is warmed at constant pressure to 90°F by coils. Find the same quantities and the relative humidity with its wet bulb for this new condition of the air. Determine the heat added to warm 1000 cu ft of the original air. Determine the heat to warm 1000 lb of original dry air and its associated moisture.

4. Find the effective temperature of air with 95°F dry bulb and 75°F wet bulb with still air. With air moving at 100 fpm, what is the effective temperature of air at 100°F dry bulb, 45°F wet bulb? What effective temperature would you suggest for a room to be occupied by persons for each of the conditions above for the first 2 hours of occupancy?

5. How much heat is produced per hour by an average person at moderate rates of work in a room of 72° dry bulb and 60% relative humidity with an air movement of 30 fpm? How is this divided between sensible and latent heat? What amount of moisture is liberated? If 50 persons working at the rate assumed are in a room circulating 1500 cfm of room air per person, what is the pick-up in moisture from the occupants? How much would this change the relative humidity of the room? Is this room too hot for a shop? By how many degrees would the sensible heat warm the circulated air? What is the equivalent volume of this circulated room air if reduced to 70°F dry?

6. If 25 kw of power were used in motors in the room of problem 5 and there were twenty 150 watt lamps in use, find the further increase in temperature of air due to these power supplies. By what methods might these temperature increases be reduced?

7. How much moisture per hour will be removed from 30,000 cfm of air at 95°F dry and 80°F wet if reduced to a saturated condition at 60°F ?

CHAPTER III

LOSS AND GAIN OF HEAT THROUGH WALLS

Heat, being a form of energy, may be measured in any unit of energy. In the United States and other English-speaking countries, it is customary to measure it in British thermal units (Btu). A British thermal unit is $\frac{1}{180}$ of the heat required to raise the temperature of 1 lb of water from 32° F to 212° F. The relation of this unit to other units of energy, as well as other dimensional relations, will be found in Table A-4.

The heat losses from rooms in cold weather or gains of heat in warm weather are due to the radiation from and conduction through the walls, doors and windows, and the so-called convection losses due to heat required to warm the leakage air in cold weather or to cool it in warm weather. In addition to this, the direct action of the sun adds heat to the rooms in both summer and winter. The heat generated in the rooms by the occupants, apparatus or machines, and by the lighting equipment, as discussed in Chapter II, is an amount which becomes a positive or negative term in problems relating to heating or air conditioning, depending on the season of the year for which the computation is made.

Heat transfer through partitions, such as building walls or the walls of radiators, is dependent on the radiation and conduction of heat, but since much of the transmitted heat is removed by fluids which bathe these partitions, convection plays an important part in the matter.

Heat transfer by radiation depends upon the difference of the fourth powers of the absolute temperatures of the bodies emitting and receiving this heat, the amount of surface of the radiating body, the reflecting or absorbing nature of the surfaces of each body and the solid angle subtended at the surface of the emitting body or source by the receiving body. This radiation from one side of a body has been reduced to the formula:

$$q_r = 0.172 EA \left[\left(\frac{T_e}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \right] \frac{\omega}{2\pi} \quad (3-1)$$

q_r = heat radiated in Btu per hour;

E = emissivity, ratio of actual emission to that of black body at a given temperature;

A = surface of source in square feet;

ω = solid angle subtended at the source by the receiver;

T_s = absolute temperature of source = $t_s + 459.6^\circ \text{ F}$;

T_r = absolute temperature of receiver = $t_r + 459.6^\circ \text{ F}$.

$$E = \frac{1}{\frac{1}{E_1} + \frac{1}{E_2} - 1} \quad (3-2)$$

E_1 refers to the heated body, E_2 to the cold body.

TABLE 3-1

VALUES OF EMISSIVITY E

Selected from Hottel as given by McAdams *

Aluminum foil . . .	0 05	Glass	0.94
Aluminum, oxidized . .	0 15	Iron and steel, polished . .	0 25
Aluminum paint . . .	0 40	Iron and steel, oxidized . .	0 85
Aluminum, polished . .	0 04	Iron and steel, rough . . .	0 95
Asbestos	0 96	Nickel, polished	0 05
Brass, dull	0 22	Oak	0.90
Brass, polished	0 03	Tin, bright	0 05
Brick	0 93	Paint	0 95
Copper, oxidized . . .	0 78	Plaster	0 91
Copper, polished	0 02	Water	0 96

* Heat Transmission, W. H. McAdams, McGraw-Hill Book Co.

For conduction the formula of heat transmission is

$$q_c = \frac{kA}{L} (t_2 - t_1) \quad (3-3)$$

q_c = heat conducted in Btu per hour;

k = coefficient of conduction or thermal conductivity in Btu per hour per square foot per degree F drop in temperature in inch of thickness;

L = thickness of conducting material in inches;

t_2 = higher temperature of material at one side in degrees F;

t_1 = lower temperature of material at L inches distance from point of temperature t_2 in degrees F;

A = surface of material conducting heat perpendicular to L in square feet.

For convection:

$$q_v = Wc(t_2 - t_1) \quad (3-4)$$

q_v = heat carried by convection in Btu per hour;

W = mass of substance heated in pounds per hour;

c = specific heat, usually specific heat at constant pressure, c_p , in Btu per pound per degree F;

t_2 = higher temperature of substance in degrees F;

t_1 = lower temperature of substance in degrees F.

Heat transfer is the name given to the heat carried per hour through a partition when expressed in terms of the temperatures of the media on the two sides of the surface, the extent of surface and a coefficient of heat transfer which takes into account the various elements of composition of the partition. It may include some radiation as well as conduction. This is given by the equation:

$$q_t = UA(t_2 - t_1) \quad (3-5)$$

q_t = heat transfer in Btu per hour;

U = coefficient of heat transfer or unit thermal transmittance in Btu per hour per square foot per degree F;

A = surface transferring heat in square feet;

t_2 = temperature of medium on hot side of partition in degrees F;

t_1 = temperature of medium on cool side of partition in degrees F.

Certain other terms are used in problems relating to heat transfer:

$$C_a = \frac{k}{L} = \text{unit conductance in Btu per unit area per hour per degree} \quad (3-6)$$

This refers to an actual thickness of the partition considered.

$$\frac{L}{k} = \text{unit thermal resistance of a substance, the resistance to heat flow in degrees per Btu for 1 sq ft for 1 hr} \quad (3-7)$$

$$\frac{L}{k} = \frac{A(t_2 - t_1)}{q_c} \quad (3-8)$$

$$f = \frac{k}{L} = \text{unit thermal conductance of an air film in Btu per square foot per hour per degree difference in temperature between the two sides of the air film} \quad (3-9)$$

$$a = \frac{1}{\frac{1}{f_1} + \frac{k_a}{L_a} + \frac{1}{f_2}} = \text{unit thermal conductance of an air space in Btu per hour per square foot per degree difference in temperature of the two surfaces forming air space} \quad (3-10)$$

$$R = \frac{1}{U} = \text{unit resistance of a partition from medium to medium in degrees per Btu for 1 sq ft for 1 hr} \quad (3-11)$$

$$R = \frac{A(t_2 - t_1)}{q_t} \quad (3-12)$$

HEAT THROUGH WALLS, PARTITIONS, ROOFS, FLOORS, WINDOWS, DOORS

When a partition or wall transmits heat from the medium on one side to the medium on the other side a temperature gradient is set up through the component parts of the wall such that the heat carried per hour per square foot by each part is the same and equal to q_t . Even if the wall is made up of one thickness of a given material, there will be films of each medium through which the heat has to pass, and in some cases the resistance of either of these films is greater than resistance of the material of the partition. In the case of the wall section, Fig. 32,

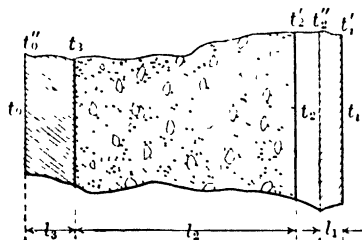


FIG. 32.—Wall Section.

the outside temperature is t_0 and the inside temperature t_1 . The wall consists of a veneer of stone of thickness L_3 , of concrete of thickness L_2 , an air space of thickness L_a and plaster of thickness L_1 as well as air films next to the surfaces. The limiting temperatures of the surfaces after steady conditions are reached are t_0'' and t_3 for the veneer, t_3 and t_2' for the concrete, t_2 for the air space and t_2'' and t_1' for the plaster. In each of these sections of wall as well as in the air films the transmission of heat per square foot per hour will be the same since there can be no change in the heat contained in any of them; all that is

received at one face must be delivered at the other. Consider one element, such as the concrete.

$$\frac{q_i}{A} = \frac{q_c}{A} = \frac{k_2}{L_2} (t_2' - t_3) \text{ or} \quad (3-13)$$

$$\frac{L_2}{k_2} = \frac{A(t_2' - t_3)}{q_i} = \text{unit thermal resistance of the concrete}$$

Similar expressions may be written for the unit thermal resistance of each part.

If now the unit thermal resistances of the different parts are added to obtain the total unit resistance of the partition, it is found, on factoring out the term A/q_i from each of the right-hand expressions, that all temperatures except t_i and t_o cancel and

$$R = \frac{L_{ai}}{k_{ai}} + \frac{L_1}{k_1} + \frac{L''_a}{k''_a} + \frac{L_a}{k_a} + \frac{L'_a}{k'_a} + \frac{L_2}{k_2} + \frac{L_3}{k_3} + \frac{L'_{ao}}{k_{ao}} = \frac{A}{q_i} (t_i - t_o) \quad (3-14)$$

The total unit resistance is equal to the left-hand side, and its reciprocal must be U since

$$q_i = UA(t_i - t_o)$$

and

$$R = \frac{1}{U} = \frac{A}{q_i} (t_i - t_o)$$

In this case the thickness of the air films L_{ai} and L'_{ao} are so difficult to determine that the quantities L/k for the films are replaced by the quantities $1/f$, and f , which equals k/L , is determined experimentally as a single term involving the effects of both factors, L and k .

The three quantities $\frac{L''_a}{k''_a} + \frac{L_a}{k_a} + \frac{L'_a}{k'_a}$ are combined in an experimental term $1/a$ as proposed by Rowley and Algren. It is then seen that the coefficient of heat transfer of a partition made up of different elements is the reciprocal of the sum of the internal resistances of separate parts.

$$U = \frac{1}{\frac{1}{f_{ai}} + \frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{1}{a} + \frac{L_3}{k_3} + \frac{1}{f_{ao}}} \quad (3-15)$$

The experimental quantity a which replaces

$$\frac{1}{\frac{L''_a}{k''_a} + \frac{L_a}{k_a} + \frac{L'_a}{k'_a}}$$

includes any radiation from one side of the air channel to the other caused by the slight temperature difference. The value of a is found to vary with the temperature of the space and the width of the space in accordance with the table below (applicable to customary building materials):

TABLE 3-2
CONDUCTANCES OF AIR SPACES a AT VARIOUS
MEAN TEMPERATURES *

Mean Temperature, Degrees Fahrenheit	Width in Inches						
	0.128	0.250	0.364	0.493	0.713	1.00	1.500
20	2.300	1.370	1.180	1.100	1.040	1.030	1.022
30	2.385	1.425	1.234	1.148	1.080	1.070	1.065
40	2.470	1.480	1.288	1.193	1.125	1.112	1.105
50	2.560	1.535	1.340	1.242	1.168	1.152	1.149
60	2.650	1.590	1.390	1.295	1.210	1.195	1.188
70	2.730	1.648	1.440	1.340	1.250	1.240	1.228
80	2.819	1.702	1.492	1.390	1.295	1.280	1.270
90	2.908	1.757	1.557	1.433	1.340	1.320	1.310
110	3.078	1.870	1.650	1.534	1.425	1.402	1.392
130	3.250	1.980	1.750	1.630	1.510	1.485	1.475
150	3.425	2.090	1.852	1.728	1.592	1.569	1.559

* From Thermal Resistance of Air Spaces, by F. B. Rowley and A. B. Algren (ASHVE Transactions, Vol. 35, 1929, by permission).

The values of $1/a$ will be increased if bright paint or aluminum foil is used to cover the surfaces of the air channel, the increase being probably 75% for the paints and 150% for foil.

Using this quantity, a , in the expression (3-15), the following is obtained:

$$U = \frac{1}{\frac{1}{f_o} + \sum \frac{L}{k} + \sum \frac{1}{a} + \frac{1}{f_i}} \quad (3-16)$$

If the value of k/L has been determined as C_a for any element of a partition then the separate factors k and L need not be used for this but merely the value C_a , and the generalized equation (3-16) becomes

$$U = \frac{1}{\frac{1}{f_o} + \sum \frac{L}{k} + \sum \frac{1}{C_a} + \sum \frac{1}{a} + \frac{1}{f_i}} \quad (3-17)$$

The quantity f for the air film (equivalent to some k/L) will vary with the nature of the surface, being less for those surfaces which hold films and permit greater thicknesses, and with the velocity of air across the surface, being larger with an increase of velocity, as this reduces the film thickness. Although the heat is assumed to be transmitted through this film by transfer, the value of f also includes the heat radiated from the hot bodies to the partition and from the partition on the other side to colder bodies. In general the value of f is determined by experiment and includes these effects, if present, since f is determined by

$$f = \frac{q_t}{A(t_1 - t_2)} \quad (3-18)$$

In this the heat q_t in Btu per hour, the surface A in square feet, and the temperatures t_1 and t_2 on the two sides of the films are measured, giving the value of f . The term q_t is inclusive and measures the heat delivered through the film by all methods of heat transmission.

Harding and Willard have found an average value of f of 1.34 for films of the room side of walls and values of 2.28 times this for the film on the outside of walls with velocities of 5 mph to 3.62 times with wind velocities of 20 mph. Rowley and his associates as well as Houghten and McDermott have investigated this question and from these results the latter investigators have suggested the following formula for value of f :

For smooth surfaces of wood, plaster or glass

$$f = 1.6 + 0.3V_w \quad (3-19)$$

For surface of concrete, stone, brick and rough plaster (medium roughness)

$$f = 2.0 + 0.4V_w \quad (3-20)$$

For rough stucco (very rough)

$$f = 2.1 + 0.5V_w \quad (3-21)$$

V_w is given in miles per hour (mph).

The heat transferred through partitions is computed by the formula

$$q_t = UA(t_i - t_o) \quad (3-22)$$

in which the value of U includes the slight radiating effect which occurs in general at the surface for ordinary temperature differences. For larger temperature differences, radiation to or from the surface must be considered separately from the film transfer.

To compute the value of U for any given type of partition, the values of k must be known, and these have been determined by experiment.

TABLE 3-3
VALUES OF k
Btu per sq ft per drop of 1° in 1 in.

Air, still.	= 0 36	Sawdust	= 0 36
Aluminum	= 1400 00	Shavings.....	= 0 40*
Brass.....	= 732 00	Slate.....	= 10 37*
Brick work.....	= 5 00†	Steel	= 400 00
Building paper....	= 0 38	Stone, typical	= 12 50*
Cement mortar . .	= 12 00†	Sandstone.....	= 10 44
Concrete, stone . .	= 12 00†	Limestone.	= 16 20
Concrete, cinder . .	= 5 20*	Fine marble	= 22 00
Copper	= 2424 00	Terra cotta.	= 6 48
Cork board.....	= 0 30†	Tile or terrazzo . . .	= 12 00†
Cotton	= 0 36	Tin	= 427 20
Felted materials . .	= 0 27†	Wood (dry)	
Fibre board . . .	= 0 33†	Maple	= 1 15†
Fill of loose material	= 0 27†	Oak	= 1 15†
Glass	= 6 48	Pine, across grain..	= 0 72
Lead	= 240 00	Pine, along grain...	= 1 32
Plaster:		Pine or fir.....	= 0 80†
Cement	= 8 00*	Light weight . . .	= 0 65*
Gypsum	= 3 30†	Woods with 16% mois-	
Stucco	= 12 00†	ture increase above	
		25%	
		Wool	= 0 29

* Taken from A S H V E Guide, 1936.

† Recommended value by Guide.

TABLE 3-4
VALUES OF C FOR GIVEN ELEMENTS

$$C_a = \frac{k}{L}$$

Btu per hour per square foot per degree F.

(From A S H V E Guide, 1935)

Flooring:		Walls:	
$\frac{1}{4}$ " battleship linoleum. . .	1 36	1" fir sheathing and paper. . .	0 71
Partition:		1" fir sheathing, paper and yel-	
Metal lath and $\frac{3}{4}$ " plaster . . .	4 40	low pine lap siding	0 50
Wood lath and $\frac{3}{4}$ " plaster....	2 50	1" fir sheathing, building paper	
Plaster board $\frac{1}{2}$ " thick.	2 82	and stucco.....	0 82
4" hollow gypsum tile	0 46	Yellow pine lap siding	1 28
Roofing:		Wall blocks:	
Asphalt composition.	6 50	Cinder blocks 8" (1 cell) . . .	0 62
Built up $\frac{3}{8}$ " thick	3 53	12" (1 cell)	0 51
Built up bitumen and felt		Concrete blocks 8" (1 cell)...	1 00
0.15" thick, 1 34 lb per sq ft,		12" (1 cell)...	0 80
with slag or gravel surface		Clay tile with air cells in direc-	
(0.1") additional, 0 83 lb		tion of heat flow:	
per sq ft.....	1 33	4" (1 cell)....	1 00
		8" (2 cells)...	0 60
		12" (3 cells)....	0 40

To explain the application of the above, the wall given in Fig. 33 will be investigated. The wall is composed of 4" of sandstone, 18" of brick work, a 2" air space, 8" of brick and 1" of smooth plaster. Where sections of the wall actually come in contact, there is no surface resistance and the wall may be considered as solid except

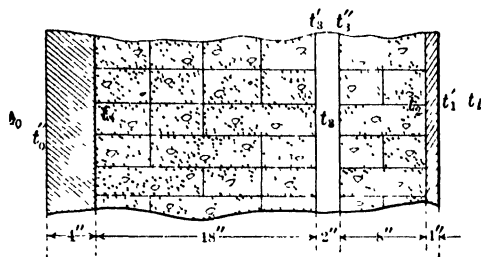


FIG. 33.—Wall Section.

for differences in the values of k for the various materials. To find f formulae (3-19-20) will be used and a will be taken from the table on page 55.

$$f_1 \text{ for smooth plaster} = 1.6 + 0.3V_w = 1.6$$

$$f_0 \text{ for stone and 15 mph wind} = 2.0 + 0.4 \times 15 = 8.0$$

$$a \text{ for 2" air space and } 50^\circ \text{ assumed} = 1.149 = 1.15$$

U is then found as follows:

$$\begin{aligned}
 U &= \frac{1}{\frac{1}{8.00} + \frac{4}{10.44} + \frac{18}{5.00} + \frac{1}{1.15} + \frac{8}{5.00} + \frac{1}{3.30} + \frac{1}{1.6}} \\
 &= \frac{1}{0.125 + 0.383 + 3.600 + 0.870 + 1.600 + 0.303 + 0.625} \\
 &= \frac{1}{7.506} = 0.133
 \end{aligned}$$

The terms of the denominator above are unit thermal resistances and show the relative value of each element of the partition as a heat insulator. The outer film offers one-fifth the unit resistance of the inner film and one-seventh the unit resistance of the interior air space. Four inches of sandstone has the unit resistance of only 2 in. of brick.

U for Wooden Floors. For a floor or ceiling as shown in A, Fig. 34, the method is quite the same. The value of U , however, must be computed for the portion of the floor covered by the joist, and then the value of U for the portion containing the air space is computed. There are two conditions to be considered for the portion including the air space. When the high temperature is at the top, there is no circulation in the air space between the plaster and the floor, and the air acts as

an insulating material. When, however, the high temperature is below, the circulation of the air transmits heat by convection and the air does not act as an insulating material as was the case just considered. In both instances, however, there are film resistances at contacts between the air and the surfaces producing temperature drops.

When the same constant, U , does not hold over a complete surface owing to changes in the construction, as occurs at studs in a partition

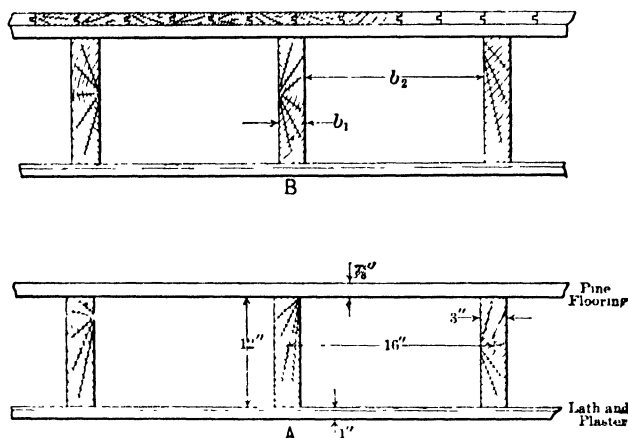


FIG 34 Wooden Floor Construction

or at joists in a floor, the average value for U to be used for the whole surface is found for these conductivities in parallel.

$$U(A_1 + A_2)(t_i - t_o) = U_1A_1(t_i - t_o) + U_2A_2(t_i - t_o)$$

$$U = \frac{U_1A_1 + U_2A_2}{A_1 + A_2} = \frac{\sum UA}{\sum A} \quad (3-23)$$

Usually the areas A have a common dimension so that the areas are proportional to the widths. If these are b_1 and b_2 there results (Fig. 34),

$$U = \frac{U_1b_1 + U_2b_2}{b_1 + b_2} \quad (3-24)$$

The mean constant for a wall is not usually found in terms of the glass and wall coefficients, the heat from each part being kept separate, but there is no reason why a mean value of U could not be found if the ratio of glass to net wall area were the same for all buildings.

It is well to note at this point that if U has been computed for a given partition, with its air films and all other parts, a value of U after

the addition of certain new parts within the partition, such as a sheet of insulating material or an increase of the thickness of some part, may be computed by finding the values of the k and L for the changed parts or the C_a , if known, or an a if a new air channel is added, and using these in the formula

$$U_{\text{new}} = \frac{1}{\frac{1}{U_{\text{old}}} + \left(\sum \frac{L}{k} + \sum \frac{1}{C_a} + \sum \frac{1}{a} \right)_{\text{new}}} \quad (3-25)$$

This formula can be applied readily to walls which differ slightly from walls for which the U has been tabulated from computation or experiment.

In Fig. 34-A, with the high temperature above, the air acts as an insulating substance, and the following results for a wooden floor:

f for smooth floor and ceiling with practically no velocity = 1.6 + 0.3 V_w = 1.6.

Computations are made for the value of U from the dimensions of Fig. 34, using lath at $\frac{3}{8}$ in. thickness and plaster $\frac{5}{8}$ in.

With the high temperature above

At joists:

$$U_j = \frac{1}{\frac{1}{1.6} + \frac{(0.875 + 12 + 0.375)}{0.72} + \frac{0.625}{3.30} + \frac{1}{1.6}} = 0.0504$$

At space between joists:

$$U_a = \frac{1}{\frac{1}{1.60} + \frac{0.875}{0.72} + \frac{2}{1.60} + \frac{12}{0.36} + \frac{0.375}{0.72} + \frac{0.625}{3.30} + \frac{1}{1.60}} = 0.0265$$

Combined

$$U = \frac{3 \times 0.0504 + 13 \times 0.0265}{16} = 0.031$$

With the high temperature below on account of the convection currents, the air does not act as an insulating substance and the following results:

$$U_j = 0.0504$$

$$U_a = \frac{1}{\frac{4}{1.60} + \frac{1.25}{0.72} + \frac{0.625}{3.30}} = 0.226;$$

$$U = \frac{3 \times 0.0504 + 13 \times 0.226}{16} = 0.193$$

If now, building paper and $\frac{7}{8}$ in. oak flooring are put on the floor, shown at A, the values of U_b for the type B, Fig. 34, become:

For hot room above

$$U = \frac{1}{\frac{1}{0.031} + \frac{0.06}{0.38} + \frac{0.875}{1.15}} = 0.030$$

For hot room below

$$U = \frac{1}{\frac{1}{0.193} + \frac{0.06}{0.38} + \frac{0.875}{1.15}} = 0.164$$

U FOR GLASS, WINDOWS AND DOORS

For glass $\frac{1}{8}$ in. thick with wind at 15 mph on outside and a circulation at 3 mph on the inside, the computed value of U would be given as follows:

$$f_o = 1.6 + 0.3 \times 15 = 6.1$$

$$f_i = 1.6 + 0.3 \times 3 = 2.5$$

$$U = \frac{1}{\frac{1}{6.1} + \frac{1}{8 \times 6.48} + \frac{1}{2.5}} = \frac{1}{0.164 + 0.019 + 0.400} = 1.71$$

For two thicknesses of glass with $\frac{1}{4}$ in. air space between at a temperature of 35° F.

$$U = \frac{1}{\frac{1}{1.71} + 0.019 + \frac{1}{1.45}} = 0.773$$

In outer air at 3 mph the value of U for single glass becomes

$$U = \frac{1}{\frac{2}{2.5} + \frac{1}{8 \times 6.48}} = \frac{1}{0.80 + 0.019} = 1.22$$

For still air on both sides of glass

$$U = \frac{1}{\frac{2}{1.6} + 0.019} = 0.788$$

If a wooden sash is used 1 in. thick the U for this wood, with 15 mph outside and 3 mph inside, would be

$$U = \frac{1}{0.164 + \frac{1}{0.72} + 0.400} = \frac{1}{0.164 + 1.39 + 0.400} = 0.512$$

For outside air, 3 mph, this becomes

$$U = \frac{1}{0.80 + 1.39} = 0.457$$

If the sash covers 15% of the window the combined U (for 15 mph outside) would be

$$\text{Window—single glass, } U = \frac{0.85 \times 1.71 + 0.15 \times 0.512}{0.85 + 0.15} = 1.53$$

$$\text{Window—double glass, } U = \frac{0.85 \times 0.773 + 0.15 \times 0.512}{0.85 + 0.15} = 0.73$$

For 3 mph outside air

$$\text{Window—single glass, } U = \frac{0.85 \times 1.22 + 0.15 \times 0.457}{0.85 + 0.15} = 1.11$$

From the above calculations it is seen that the values of U for glass windows vary with velocities and also with the amount of sash and frame. Experiments have been made by Moyer and his associates, by Allen, by Willard and Lichty, and by Rowley and his associates on transmission through glass and sash. From these, the values of U recommended by the Guide of the A.S.H.V.E. for windows and skylights (based on values of Harding and Willard with wind at 15 mph) are as follows:

TABLE 3-5
VALUES OF U FOR WINDOWS AND SKYLIGHTS

(Area includes frame)

Single glass	. . .	1.13
Double glass	. . .	0.45
Triple glass	0.28

These values are recommended, although smaller than computed values given above, as they represent the values employed in practice by the profession.

For doorways with thin doors, the same value, 1.13, is suggested by the Guide as the value of U for doors, although 0.57 has been shown to be the value of U for a 1 in. outside wooden door. The author has preferred to use for the U for doorways the U for windows because of the leakage of heat which occurs on opening the doors. If the wood were made 2 in. thick in the above calculation, the values of U would be changed to 0.42 for an outside door, and where doors are closed for long periods these lower values should be used.

The value of the unit thermal transmittance U for definite structures may be tabulated, and values have been computed by the author and compared with those given in the Guide. These latter are the results of many experiments performed on actual wall sections and have been reported in the Transactions of the A.S.H.V.E.

The constructions used in practice will be described at first, after which values of U for these peculiar structures will be tabulated in such a way that values for any type with variations of dimensions may be readily selected.

Walls. For many buildings the permissible types of walls, as well as other details, are fixed by the building code of the city in which the

structure is to be erected, and the engineer has no free choice of construction. He must use the types fixed by law, from which he may make his selection. Frequently walls are built of brick. Bricks vary in size in different localities, but a standard of $8\frac{1}{4} \times 4 \times 2\frac{1}{2}$ in. has been adopted by the National Brickmakers' Association. Walls are built in various thicknesses, and the successive layers of brick, both horizontally and vertically, are laid with broken joints. This means that there is no straight joint for more than one brick through which water might work its way, or leakage of air might occur. The bricks laid lengthwise are called stretchers, and those with length perpendicular

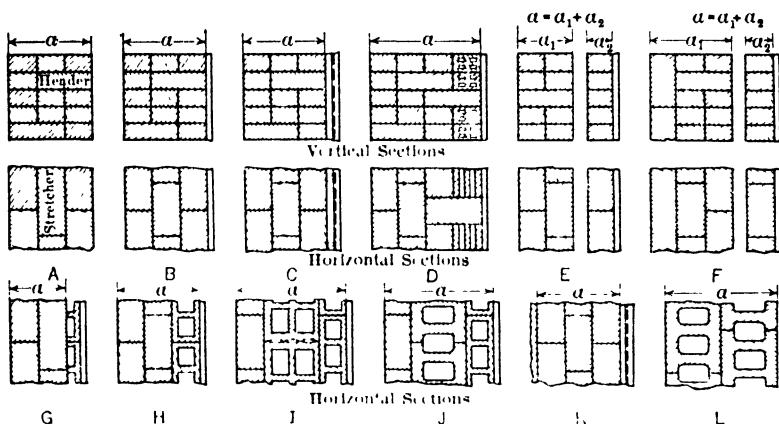


FIG. 35 Masonry Wall Sections.

to wall are called headers. After six or seven stretcher courses, bricks are laid as headers to bond the separate tiers or vertical layers. This is shown at *A*, Fig. 35.

At times an air space is formed in the wall, as shown in *E*, Fig. 35, so that the interior of the wall will be dry and will be a better non-conductor of heat. When a solid wall is built it is not advisable to place the plaster directly on the wall, as in *B*, Fig. 35, because the water soaking through the wall will produce a damp surface. To prevent this, the wall *C*, Fig. 35, is furred before the plaster is put on. This consists in fastening wood or iron furring strips at 16 in. centers on the wall and plastering upon the wood or metal lath attached to the strips. The wooden furring strips are usually $\frac{3}{4} \times 2$ in. so that the laths are separated from the wall by almost $\frac{3}{4}$ in., thus forming an insulating air space. The furring strips are attached to wooden wedges or plugs driven into the joints of the walls. The same effect is obtained by the use of hollow bricks as the lining of the wall. These are shown

at *D*, Fig. 35. At times 2 or 4 in. hollow tile are used behind the brick in place of furring for the same purpose. Such walls are shown in Fig. 35, *G* and *H*, while *I* represents the type of wall in which hollow tile is used for the wall and a 4 in. veneer of brick for facing. It is advisable that more than one tile be used for the thickness of these walls. Hollow concrete blocks of stone or cinder concrete are shown

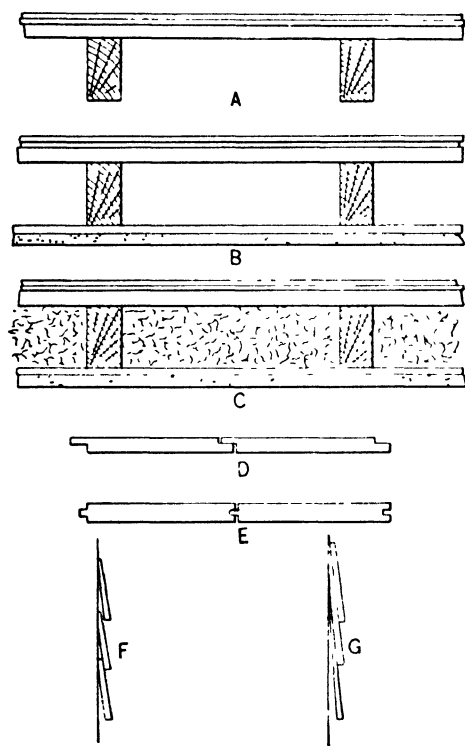


FIG. 36.—Wooden Wall Construction.

members of these are generally of 2 × 4 in., 3 × 4 in. or 4 × 6 in. timbers placed at 16 in. centers, and on these sheathing boards are nailed. These are boards planed at least on one side (surfaced) to bring them to a uniform thickness. Shiplap, as shown in *D*, Fig. 36, or tongued and grooved boards, *E*, Fig. 36, serve to make tighter joints for the purpose of keeping down the leakage of air. Building paper of one or two thicknesses is placed on top of the sheathing, and after this, clapboards, *F*, or shingles, *G*, are placed on top of the paper. On account of the frequent vertical joints in the shingles, these are usually placed so that not more than one-third of the length of the

at *J*. In block walls as well as brick walls it is advisable to break joints at different layers and tiers. The use of two thicknesses with broken joints makes a tighter wall for heat, rain or air leakage. Single block walls are always objectionable. A rough brick wall covered with stucco and plastered on the inside with furred lath is shown at *K*, Fig. 35, and *L* indicates a wall of concrete blocks without finish on either side.

Stone and concrete walls are used for buildings, and constants have been computed for these. At *F*, Fig. 35, the facing or veneer of cut stone appears on a brick wall, and this same form of facing is found on rough masonry walls.

Wooden walls or frame constructions are used at times. The studs or upright

shingle is exposed to the weather. In width, shingles must break joints with the joints of the layer beneath.

A wooden wall, without plaster, is shown at A, Fig. 36, while B and C are plastered on the interior, the space between studs of C being filled with a loose material, such as rock wool, asbestos wool or even sawdust or shavings. In wooden construction of planed or surfaced lumber the nominal sizes are usually greater than the actual dimension by $\frac{1}{4}$ in. for single surfacing or $\frac{3}{8}$ in. for planing on two opposite sides. The actual dimensions must be known in computing the heat transfer coefficients.

Walls with Bright Metal Insulation. Walls have been constructed of sheets of polished metal forming air channels, the resistance to heat flow being due to the films and the small emissivity of such surfaces. Queer * at the Experiment Station of Pennsylvania State College has shown that in a 1 in. horizontal air space with black metal surfaces the conductance C of the channel is 1.095 Btu per sq ft per hr per deg difference in temperature when the mean temperature is 50° F, and 1.465 Btu per sq ft per hr per deg difference when the mean temperature is 100° F. If a thin, bright aluminum foil is introduced to reduce these spaces to two $\frac{1}{2}$ in. channels, these conductances are reduced to 0.245 and 0.310 respectively. When the original metal surfaces were covered with aluminum foil in this channel of 1 in. thickness the values of the conductances were 0.43 and 0.59 respectively at 50° and 100° mean temperature.

On making tests with vertical air channels the conductance, C , was found to decrease with the increase in width of air space up to $\frac{3}{4}$ in. and also with the height of the free air column up to about 35 in. The maximum air space thickness recommended by Queer is $\frac{3}{4}$ in. for single spaces and $\frac{3}{8}$ in. for multiple spaces.

The formula for such walls to care for diffusion (conduction and convection) as the first term and radiation as second is

$$q_t = \frac{k_w k_h \Delta T^{1.226}}{(n+1)^{1.1}} + \frac{0.172}{n+1} \times \frac{E_1 E_2}{E_1 + E_2 - E_1 E_2} \left[\left(\frac{T_1}{100} \right)^4 - \left(\frac{T_2}{100} \right)^4 \right] \quad (3-26)$$

k_w = air space width factor, 0.60 for $\frac{3}{8}$ in., 0.52 for $\frac{1}{2}$ in.
and 0.47 for $\frac{3}{4}$ in.;

k_h = air space height factor, 0.25 for 35 in.;

$\Delta T = T_1 - T_2$;

* Importance of Radiation in Heat Transfer through Air Space, by E. R. Queer, Trans. A.S.H.V.E., 38-1932.

T_1 and T_2 = absolute temperatures of two sides;

n = number of screens;

E_1 = 0.065 for aluminum at 50° mean temperature, 0.0685
for aluminum at 100° mean temperature;

E_2 = emissivity of other surfaces. See page 51.

Somewhat similar results of experiments by Rowley * of Minnesota have been reported, as well as some by Dufton † from the National Physical Laboratory of Great Britain.

Several loose crumpled sheets of thin, bright aluminum foil, as used by the Alfol Insulating Company, are employed in air channels of walls or in pipe or surface coverings as insulators.

Floors. Wooden floors for dwellings and many small buildings are built as shown in Fig. 34. In this form, wooden joists 2×10 in., 2×12 in. or 3×12 in. are placed on 16 in. centers with a cover or floor composed of one or two layers of tongued and grooved floor boards or a lower layer of shiplap or square-edged boards with the tongued and grooved boards in the upper layer separated by building paper. In all cases at least one side of the boards must be planed to bring them to a uniform thickness. To the lower side of the joists, laths are attached when the lower surface is to be plastered.

In mill or slow-burning construction, Fig. 37, heavy wooden girders (12×12 in.) are placed at about 8 ft centers, on which is a floor made of 3 in. planks. The planks are usually of yellow pine, and above them maple boards may be placed, separated from the planks by building paper of some kind.

Fig. 38 shows floor construction, using reinforced, broken stone or cinder concrete slabs between I-beams. The concrete encases the beams. The ceiling below is carried on metal laths attached to small channel irons running from beam to beam, if a flush ceiling is desired, or the plaster is attached directly to the concrete if a panel effect is desired. Often the concrete is placed with such care, using smooth sheet forms, that no plaster is needed. On top of the reinforced concrete slab, for a wooden floor, a lean cinder concrete fill is made between the dovetailed sleepers to which the floor boards are attached. The sleepers usually run at right angles to the beams. Two thicknesses of floor boards with paper between are used, the upper surface being

* Insulation Value of Bright Metallic Surfaces, by F. B. Rowley, Heating, Piping and Air Conditioning, June, 1934.

† Metallic Insulation, by A. F. Dufton, Heating, Piping and Air Conditioning, February, 1934.

of maple when lasting qualities are desired. When a masonry floor is required a cement and sand mixture with smooth finish, or a terrazzo layer, is placed on the concrete slab. At times the ceiling laths are suspended some distance from the beams to secure space for concealed equipment. This is known as a hung ceiling.

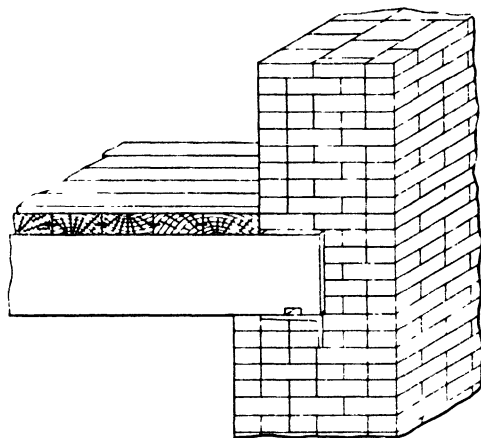


FIG. 37.—Mill or Slow Burning Construction.

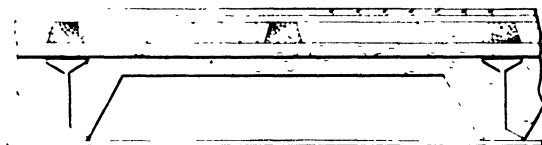


FIG. 38.—Reinforced Concrete Floor.

Roofs. Roofs are built in many ways, depending on the structure. Some roofs are built of reinforced concrete, as shown in Fig. 39. On top of the reinforced concrete slab, which may be of broken stone or cinder as the aggregate, a fill of cinder with enough cement and sand to give a slight bond is spread to a definite grade so as to guide the rain water to the catch basin. On top of this is a screed coat as a base for water-tight roofing material, composed of four layers of roofing felt cemented together by hot bitumen or pitch. Often on top of this is a covering of broken slag or gravel for protection, but when the roof is to be used for a promenade, flat ceramic tile is set in cement mortar on top of the waterproofing membrane. If there is no air space between the roof and the room beneath, it is customary to introduce a layer of heat-insulating material like the cork board shown.

Fig. 40 shows a special portable reinforced slab, used for roofs, the size of which is so related to the span between roof beams that it can be spaced readily and have sufficient strength.

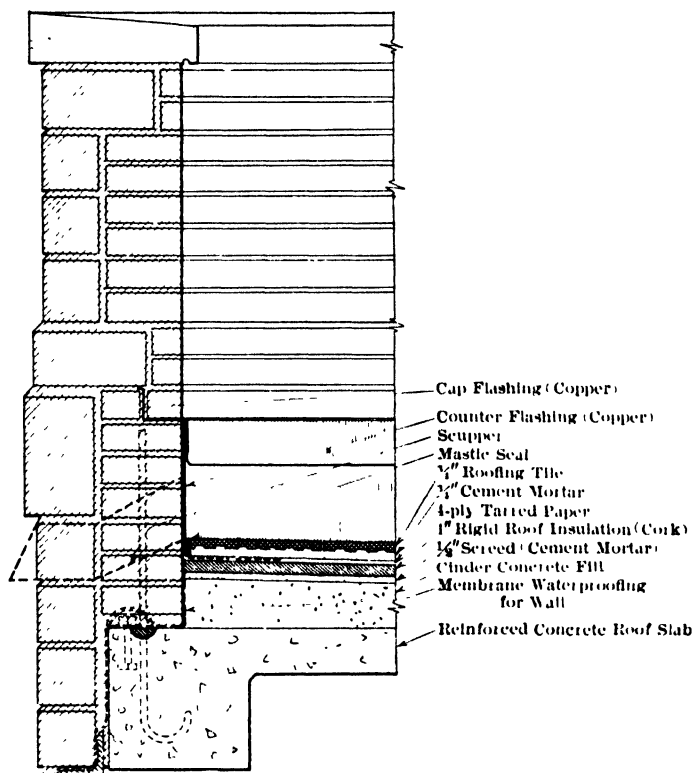


FIG. 39—Roof Construction.

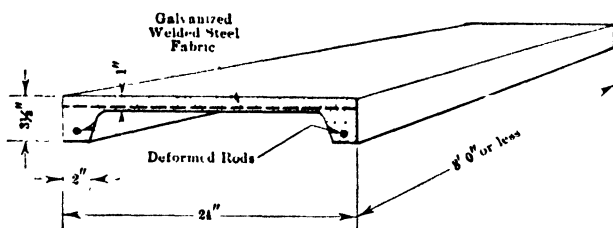


FIG. 40.—Preformed Roof Slab.

For inclined roofs, the reinforced deck, Fig. 41, may be replaced by special tiles 3 in. \times 12 in. \times 15 in. which are supported by T-iron purlins attached to the main roof beams. Two special forms of roof

tile are shown, and in this figure a hung ceiling is indicated. This forms an air space beneath the roof which is of value in cutting down heat loss.

When metal roof coverings of copper are to be used, a wooden surface is formed on top of the surfaces described above.

For frame houses the roof is formed like the walls of the house shown in Fig. 36. The roof rafters are usually 2×8 in. or 2×10 in.

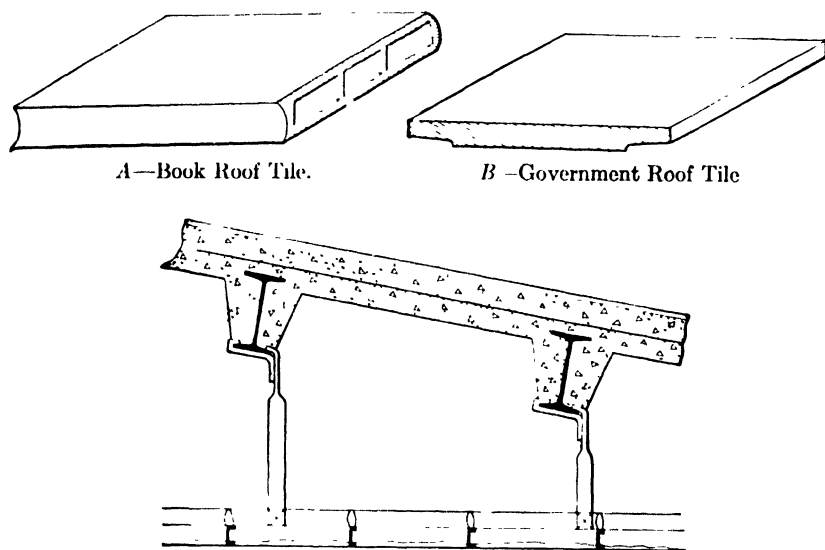


FIG. 41.-Roof with Hung Ceiling-Roof Tile.

and are placed on 16 in. centers. These carry the roof sheathing, on top of which is placed tin or copper for a metal roof, or building paper and shingles or slates if these are used for the outer layer. The form *B*, Fig. 36, may indicate plaster on the under side of the roof purlins. This, or some form of composition plaster board, should be attached to the lower side of the rafters to increase the insulating qualities of the roof, and if the method of *C*, Fig. 36, is followed, the loose filling in the space between purlins greatly reduces the heat loss in the winter and the heat gain in the summer.

Partitions. The horizontal sections of the ordinary forms of partitions are shown in Fig. 42. *A* represents the standard form of wood partition made of 2×4 in. or of 3×4 in. (actually $\frac{3}{8}$ in. less in each direction because of finish on four sides) studs of spruce, hemlock or fir placed on 16 in. centers to which are fastened wood or metal laths

or a patented sheeted substance known as plaster board, over which plaster is placed. Wooden laths are $\frac{3}{8}$ in. thick and covered with $\frac{5}{8}$ in. plaster, while the plaster thickness on metal lath is $\frac{3}{8}$ in. The plaster board is $\frac{3}{8}$ in. thick and carries $\frac{1}{2}$ in. of plaster.

Form *B* consists of a single row of hollow tile with plaster on each side, or it may represent light-weight gypsum tile of larger size with plaster; and *C* shows two sets of tile with an air space between them. *E* illustrates a partition formed of wire lath attached to channel irons, and *D* is one in which expanded metal supports the plaster while the space on the inside may be filled with asbestos to deaden the sound as well as to cut down air currents and thus reduce heat transmission.

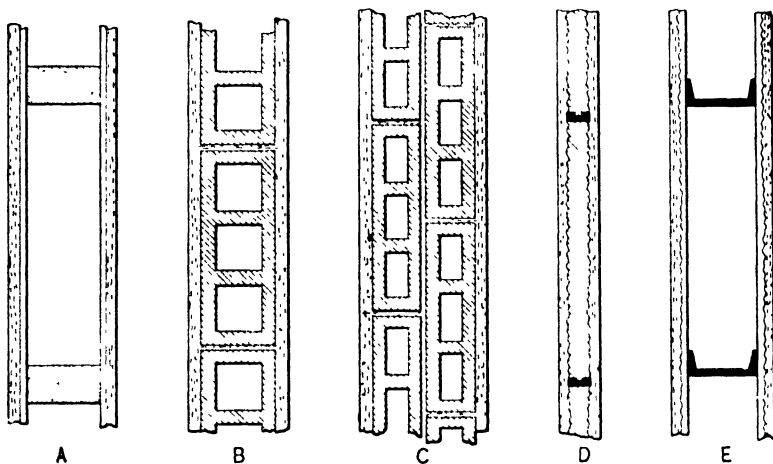
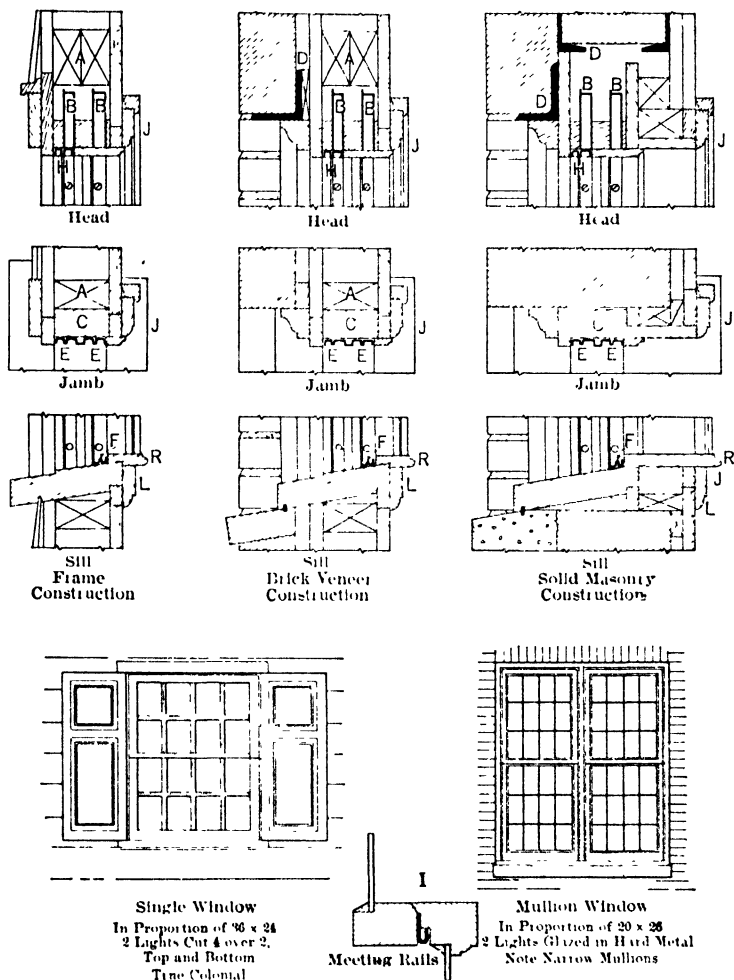


FIG. 42.—Partitions.

Windows and doors are constructed with a frame within which the moving sashes or doors are mounted. The frame consists of a head or top horizontal member, two side members, called jambs, and a lower horizontal member known as a sill. In door frames the sill is often omitted as the door jambs are supported by the wall or partition on the side and at the bottom by the floor or a separate sill.

Windows built with two sashes which move vertically in grooves are known as double-hung windows. Such windows built of wood are shown in Fig. 43. In this figure the section at the head, jamb and sill is shown for application to frame construction, to brick veneer wall over frame and to solid masonry construction. The parts marked *A* represent the members to which the frame is attached as the wall is being built, or after completion of the framing or masonry, to which the window frame may be attached. The parts *B* of these particular

frames represent the location of the patented balances placed above the head to support the sash when raised and aid in lifting them. Iron or lead sash weights which travel in channels built into the jambs *C* may be used for balancing. They are connected to the sash by sash cords



Courtesy of N. S. W. Company

FIG. 43.—Double Hung Windows.

of special weaving or by chains which pass over sash pulleys built into the jamb at the top of each sash groove. The angle-iron lintels, *D*, carry the masonry above the window opening. Grooves in the side members of the sash receive tongues of metal, *E*, attached to the run-

ways of the frame to form weather stripping which prevents air leakage. The tongue, *F*, in the sill prevents air leakage at the bottom rail of the inner sash, and tongue *H* cares for the top rail of the outer sash. The upper and lower sash are provided with interlocking metal strips, as shown at *I*, to accomplish this sealing at the center of the window. In this way the cracks in the two sides for the full height of the window, and in the top, middle and bottom for the width of the window, are made so small that leakage is reduced. A locking device on the two

sashes at their meeting line draws them together and reduces air leakage.

To cut down leakage around a frame the space between the frame and the building is calked with oakum or some bitumastic material. The parts *J* shown around the window are called the trim, *R* being the sill and *L* the apron.

In many modern buildings the frame and sash of double-hung windows are made of metal. These forms may be seen in such publications as Sweets' Catalogue File. In installing metal frames care must be exercised to seal or

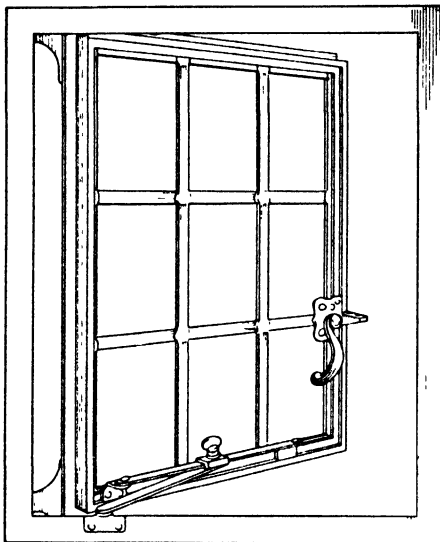


FIG. 44.—Metal Casement Window.

calk the space between the frame and the wall.

Casement windows, Fig. 44, are those built with hinged sash, which may be made of wood although an all-metal window is shown. These are arranged to open either outward or inward, and manufacturers build combinations of movable and fixed panels suitable for different widths and heights of window openings. For factories, windows pivoted on horizontal trunnions at the middle of the sash, Fig. 45, are used at times as well as those with the trunnions at top or bottom of the sash. Here, again, the manufacturers have many standard combinations to suit different sizes of window openings, and the architect designs his opening to fit one of these standard sizes.

Doors. The arrangement of a metal door frame and wood door with metal face ("Kalamein" door) is shown in Fig. 46. In erection, the metal-door frame is placed in position and then the partition blocks or

partition studs are built into the space in the jambs and head so that it is firmly anchored. The frame is formed with the door stop, *A*, on the jambs and head, and the door, attached to this frame by hinges, comes against these stops when closed, being held in this position by a knob-controlled latch or the bolt of a lock. The lower part of the door opening, known as the threshold, is formed by the floor of the building, or at times a saddle is placed at this point to make a better seal. Wooden door frames are used at times. These receive separate trim on the wall face around the exterior, as shown for the window of Fig. 43. This trim in Fig. 46 is part of the frame.

The above figures represent various typical methods of building construction, and the transmission coefficients which are computed for them will serve as guides for other constructions if time is not available for computing U . Any element added to those indi-

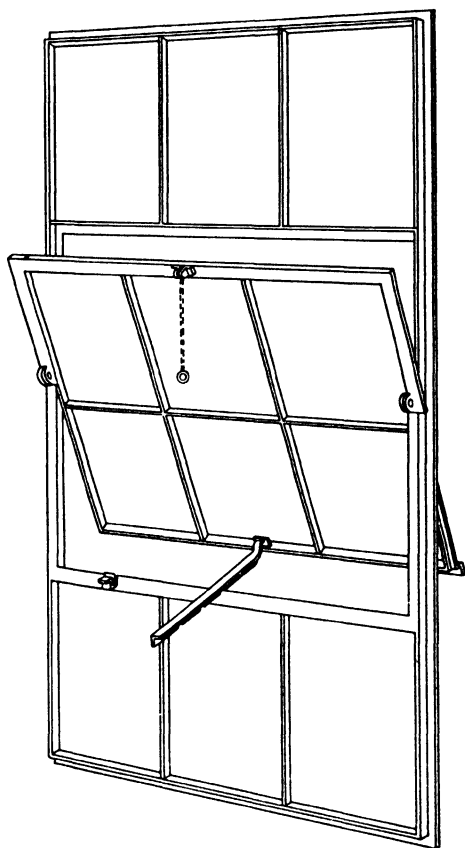


FIG. 45.—Pivoted Window.

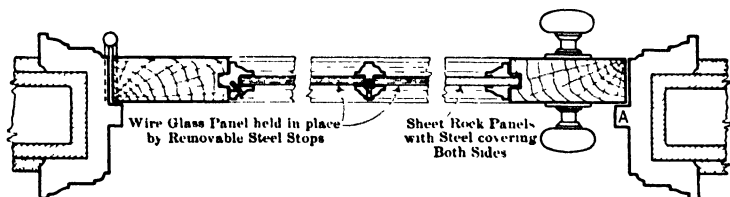


FIG. 46.—Metal Door Frame and Kalamein Door.

cated, or any change in thickness of a particular part, may be computed by:

$$U_{\text{new}} = \frac{1}{\frac{1}{U_{\text{old}}} + \frac{L}{K_{\text{new}}}} \quad \text{or} \quad \frac{1}{\frac{1}{U_{\text{old}}} + \frac{1}{C_{\text{new}}}} \quad (3-27)$$

TABLE 3-6
VALUES OF U
WALLS (Fig. 35)
For Dimension a

a	A	B	C	D	E	F	G	H	I	J
8"	0 42	0 40	0 29	0 34	0 29	0 30	0 35
12"	0 32	0 30	0 24	0 27	0 24	.	0 24	0 29
16"	0 25	0 24	0 20	0 22	0 20	0 22	0 20	0 24	0 22	0 24
20"	0 21	0 20	0 17	0 19	0 17	0 19	0 18	0 20	0 19	0 20
24"	0 18	0 18	0 15	0 16	0 15	0 16	0 15	0 17

a	K	L	Stone Masonry			Concrete			Wooden. Fig. 36
			A	B	C	A	B	C	
8"	0 29	0 43	0 72	0 65	0 41	0 71	0 64	0 41	Fig. A 0 29
12"	0 24	.	0 58	0 53	0 36	0 57	0 53	0 36	Fig. B 0 20
16"	0 20	0 26	0 49	0 46	0 32	0 48	0 45	0 32	Fig. C 0 06
20"	0 17	.	0 42	0 40	0 29	0 41	0 39	0 29	Aluminum
24"	0 15	.	0 37	0 35	0 27	0 36	0 35	0 26	Three $\frac{1}{2}$ " spaces 0 072

PARTITIONS (Fig. 12)

Fig. A, Plaster one side . . .	0 46	Fig. C, Plaster two sides .	0 25
Fig. A, " " two sides. . .	0 26	Fig. D, " " " "	0 14
Fig. B, " " " "	0 10	Fig. E, " " " "	0 39

FLOORS AND CEILINGS

Type	Heated Room Above	Heated Room Below	Type	Heated Room Above	Heated Room Below
Fig. 34 A, with plaster.	0 03	0 19	Fig. 37	0 14	0 14
A, without plaster	0 34	0 34	Fig. 38	0 07	0 19
B, with plaster . .	0 03	0 16	Fig. 38, no ceiling	0 25	0 25
B, without plaster	0 27	0 27			

ROOFS

Type		Type	
Fig. 41, with ceiling	0 22	Shingle, without sheathing .	0 49
Fig. 41, without ceiling	0 42	Shingle, with sheathing .	0 30
Book tile and wood . .	0 30	Shingle, sheathing, plaster	0 19
Government tile and wood .	0 39	Fig. 39.	0 16

WINDOWS AND DOORS

			Outside	Inside
Single glass	1 13	$\frac{3}{4}$ " door	0 69	0 52
Double glass	0 45	1" door	0 57	0 45
Triple glass	0 28	$1\frac{1}{4}$ " door	0 48	0 39
Quadruple glass	0 20	2" door	0 42	0 34

After the values of U to use for all parts of the room are determined, the next quantities to find are the areas to which these values of U refer. The areas A of the conducting and radiating walls or partitions, as well as the areas of the floor and ceiling, are found from the plans of the building. The total area of each face of the room is found; the values are kept separate and called north wall, east wall, south wall, west wall, floor and ceiling. The glass and door areas of each of the walls are found, using the size of the opening filled by their frames. The sum of these is subtracted from the gross area of its own face, and the difference is known as net face area. Door area is considered to be equal to window area for heat transfer because of the frequent opening and closing of the door.

Sun Effect. The effect of radiant energy of the sun has been investigated by Houghten and his associates * of the Research Laboratory of the A.S.H.V.E. in the Pittsburgh Station of the U. S. Bureau of Mines, using eight roof panels of different roof construction: concrete, gypsum, wood, cork, insulated concrete and laminated iron and cork. They measured the radiant energy with a pyrheliometer, obtaining 311 Btu per hr per sq ft of surface normal to the sun's rays at noon on a clear day; the Weather Bureau has reported 338 Btu per sq ft per hr. The effect on horizontal and vertical surfaces will equal this, multiplied by the sine of the angle of inclination of the rays to the surface at any instant, so that, although the south wall receives the noon-day sun, the latitude of most of the United States is such that the sun effect per square foot of south wall at noon during July is given by these experimenters as about 100 Btu per sq ft per hr, whereas an east wall at 8 A.M. or a west wall at 4 P.M. receives 210 Btu per sq ft per hr. The roof in the latitude of Pittsburgh may receive as much as 295 Btu per sq ft per hr. The hourly value of these rates varies with the inclination of the sun at different hours of the day, forming a modified sine curve.

As the sun strikes a surface, some of the energy is reflected and that which is absorbed increases the temperature of the wall or roof, causing heat conduction through the structure, radiation back to outside bodies of lower temperature and even conduction through the outer air film to the surrounding air. As the intensity of the sun's energy decreases the outer temperature falls and there will be an out-

* Studies of Solar Radiation through Bare and Shaded Windows, by Houghten, Gutberlet and Blackshaw, Heating, Piping and Air Conditioning, 1934.

Absorption of Solar Radiation in Relation to Temperature, Color, Angles and Other Characteristics, by Houghten and Gutberlet, Trans. A.S.H.V.E., Vol. 36, 1930.

Heat Transmission as Influenced by Heat Capacity and Solar Radiation, by Houghten, Blackshaw, Pugh and McDermott, A.S.H.V.E., Trans., Vol. 38, 1932.

ward flow of heat when the outer surface reaches a temperature below that attained by the inner parts.

The report of the tests gives the variations of the temperatures of the top surface of the roof, of the bottom surface, of the outside air and of the inside air. Although the inside temperatures were kept practically constant during 24 hours there was great variation in the temperature of the top surface (50° to 160° F) and of the outside air (55° to 90° F). The temperature of the bottom surface of the roof varied about 10 degrees. Once it was lower than the top surface during the day from about 8 A.M. and higher at night from about 8 P.M. Sometimes it even became less than the inside temperature. The Nicholls heat flow meters * placed in the top and bottom surfaces of the slabs showed a flow into the roof surface during the daytime and a return of heat to the outside air at night, while the flow from the bottom meter into the space around it varied a little and became slightly negative in some cases, depending on the type of panel. As heat capacity of the material of the wall or roof becomes greater this will delay the transmission of heat to the interior of the building and also the time at which it reaches a temperature below that of the interior so as to absorb heat from it, assuming that the temperature of the outer air falls sufficiently below that of the building to cause this.

The maximum heat flow occurred after the maximum temperature of the outer surface, slightly after the noon hour.

The average maximum rates of heat flow into the building and ranges of time lag between maximum temperature and maximum heat flow are given in the table:

TABLE 3-7
HEAT FLOW THROUGH ROOF STRUCTURE

Roof Material	Max Rate, Btu per hr per sq ft	Lag	K	f _i	U	Equiv ΔT
2" Laminated iron and cork	9 7	0.9 hr-3.7 hr	0.417	1.0	0.17	57
4" Laminated iron and cork	6 2	5 hr-9.5 hr	0.754	1.25	0.15	41
8" Laminated iron and cork	6 0	15 hr-24 hr	1.231	1.25	0.13	46
2" Pine	25 0	1 hr-2 hr	0.830	1.9	0.31	81
6" Concrete	32.0	2 hr-3.8 hr	8.000	1.9	0.65	50
4" Gypsum	14 1	1.7 hr-3.3 hr	1.445	1.9	0.28	50
4" Cork-concrete	10 9	2.3 hr-3.6 hr	1.218	1.9	0.25	44
4" Concrete-cork . . .	16 5	1.16 hr-2.16 hr	1.218	1.9	0.25	66

* Heat Flow Meters, by Nicholls, Trans. A.S.H.V.E., 30, 1924.

Using average values of K and f , given by the experimenters, and a value of f_0 , the values of U have been computed. Then the equivalent temperature difference to care for the maximum heat flow with this U is determined. The average value of these is 50°F if the value from the wooden roof is omitted.

To simulate the various types of roof and wall surfaces, the experimenters painted the black oilcloth covering with lampblack, red brick dust and aluminum bronze and found that the heat absorbed was reduced respectively to 94%, 63.4% and 28.2% of their previous values with black oilcloth.

These percentages applied to the 50° rise from the experiments give approximately the increase of temperature that is allowed by the A.S.H.V.E. Guide for computing the sun effect on horizontal roofs. For walls, these temperature increases for roofs are multiplied by 100/295 for the south walls and 210/295 for east and west walls.

These ratios give the following results, which are in close agreement with the allowances of the Guide.

TABLE 3-8
TEMPERATURE DIFFERENCE TO BE ADDED FOR SUN EFFECT

Surface	Color		
	Black	Red	Aluminum
Roof, horizontal	47	32	14
East or west wall	33	23	10
South wall	16	11	5

$$\text{Heat from sun effect} = UA (\text{Temperature Difference}) \quad (3-28)$$

The large sun effect on the roof indicates the advisability of an air space at this point, and the ventilation of this space or an attic by a fan in summer to prevent heat entering the rooms on the top floor as well as cooling of the space with night air.

Glass was shown to reduce the intensity of the normal solar radiation by about $12\frac{1}{2}\%$ and the remainder was transmitted to the room, but because of the opacity of glass to the short wave lengths of radiation from bodies within the room this heat is not reflected back from the room. That absorbed by the glass at angles differing from 90° is also radiated and convected to the medium on each side so that about 97% of the sun radiation is delivered to the room. If the direction of the

office buildings, Walker has recommended that a cooling system be so zoned that its output may be altered as the sun moves from the eastern to the western sides of the building.

The time lag is such that, in a room having eastern and southern exposure (in northern latitudes), the eastern wall will be delivering its heat when the southern glass is delivering its heat, and these should be added. The maximum effect from the roof may occur at the time of maximum glass effect on the west side, and the maximum effect from the south wall may occur when the glass effect on the south wall has been reduced to one-half of its maximum value. These time lags must be considered in getting the actual load on the system.

The sun effect on glass, walls and roof does diminish the heat demand during the heating season when the sun shines, but on cloudy days and at night the heating system must supply the full heat losses, and hence the sun effect is not used in computing the heat demand for the system. With air-conditioning systems, however, for the warm months, the same thing is true about cloudy days and nights, but the system must be large enough to remove the heat from the sun whenever this is present, hence the necessity of its determination and use in computing the cooling load.

Temperature of Rooms and Atmosphere. In figuring t_i and t_o , the temperature on the two sides of a wall, it is well to remember that the use for which the room is intended fixes one of these and, if a wall is an outside one, the other, t_o , is fixed by the author as the lowest temperature which may continue for several days or for a week. In the latitude of northern New York t_o may be taken as 0° F , or it may be -10° F ; in New York City 10° F or 20° F may be used. In Washington 20° F or 30° F might be used. The A.S.H.V.E. Guide suggests the use of a temperature 15° above the lowest temperature reported during the last ten years for the region in which the structure is located, and for massive buildings it suggests an even greater difference than 15° above this minimum. It reports minima values of temperature, direction and value of prevailing wind in a complete table for cities in all parts of the United States and Canada. This is referred to for extended information. However, the following average values of minimum temperature have been arranged from this table by the author for different regions:

New England and New York	-23° F ($-35^\circ \text{ N. H. to } -15^\circ \text{ Conn.}$)
Middle Atlantic Coast States (N. J., Pa., Del., Md., Va.)	-8° F ($-13^\circ \text{ Pa. to } -3^\circ \text{ Va.}$)
South Atlantic Coast States (N. C., S. C., Ga., Fla.)	$+ 4^\circ \text{ F}$ ($0^\circ \text{ Ga. to } 10^\circ \text{ Fla.}$)
Gulf States	$- 2^\circ \text{ F}$ ($-6^\circ \text{ Ala. to } +1^\circ \text{ La.}$)
(Ala., Miss., La., Texas)	

Central Belt States	-21° F (-28° W. Va. to -12° Tenn.) (W. Va., Ky., Tenn., O., Ind., Ill., Mich.)
Northern Plain States	-38° F (-44° N. Dak. to -32° Neb.) (Wis., Minn., S. Dak., Ia., Neb., N. Dak.)
Southern Plain States	-21° F (-25° Mo. to -14° Ark.) (Mo., Ark., Okla., Kan.)
North Mountain States	-36° F (-53° Mont. to -25° Idaho) (Mont., Wyo., Colo., Idaho)
Southern Mountain States	-15° F (-22° Utah to -6° Ariz.) (Utah, Nev., Ariz., New Mex.)
North Pacific States	-13° (-13° Ore., and -13° Wash.) (Ore. and Wash.)
South Pacific State	+27° F (27° Cal.) (Cal.)

In using these for a given city, the direction of the winter wind should be ascertained.

Some specifications formerly called for sufficient radiation to heat the building to a desired temperature in zero weather, but this temperature should not be used in design unless zero weather is found for several consecutive days in the locality considered. The specifications should mention the actual outside temperature fixed by the region in which the structure is to be built, if a guarantee is required. **The designer should not ask for a guarantee if he has specified the amount of radiation with the size of its attached boiler. The responsibility is his and not that of the contractor.**

The temperature for rooms for various purposes, as given in Chapter II, for comfort or use are the temperatures found about 5 ft from the floor (breathing line) and may be used as the average t_i for rooms which are not more than 10 ft high. Some designers prefer to use the temperature at the so-called comfort level of 2 ft 6 in. When the room is higher than 10 ft, the average temperature t_i is higher than the values at the breathing line. According to Rietschel, if the value desired at head height is t , the mean temperature t_i is given by

$$t_i = t + 0.017(h - 10)t \quad (3-29)$$

t_i never exceeds 1.15 t .

t = desired temperature head high (breathing line);

h = height of room in feet;

t_i = mean temperature;

If t_e = temperature at ceiling.

$$t_e = t + 0.035(h - 10)t \text{ (not over 1.30}t\text{)} \quad (3-30)$$

This temperature t_c should be used in computing the heat transfer of the ceiling.

The A.S.H.V.E. Guide recommends that the temperature at the floor of the rooms at ground level or above unheated rooms be taken as 5° less than temperature at 5 ft level provided the temperature at the breathing line is not less than 55° F, as tests show that this difference does occur.

When a room is not heated, the heat transfer from the surrounding rooms gives heat sufficient to hold it at a temperature above that of the outside air.

As an example of such spaces, the temperature of attics may be computed by equating the heat loss through the roof and walls to the heat gain from the floor of the attic. This gives

$$U_F A_F (t_i - t_a) = U_r A_r (t_a - t_o) + U_w A_w (t_a - t_o)$$

where F refers to the coefficient of the floor of the attic, w to the wall and r to the roof. The only unknown is t_a , and this may be readily computed. The author would recommend the following if sufficient data are not at hand to determine the temperature:

TABLE 3-10
TEMPERATURES OF UNHEATED SPACES

Cellars	36° F
Vestibules	20° F
Attic under slate roof	25° F
Attic under booktile and metal roof	40° F
Attic under wood and metal roof	32° F

The A.S.H.V.E. Guide recommends that the temperature of the unheated space should be taken as the arithmetic mean of the outside temperature and the average of the temperatures of the heated rooms surrounding this unheated space.

The mean temperatures of the walls of each room or space in a building as well as the floor and ceiling temperatures of high rooms having been found, these temperatures should be marked on the plans in each room.

Effect of Exposure and Intermittent Heat. The values of U have been computed for wind velocities of 15 mph, and if such velocities are never found in the coldest winter month, a correction can readily be made by the values of the film factor for 15 mph and for the actual wind velocity.

$$U_{\text{new}} = \frac{1}{\frac{1}{U_{15}} - \frac{1}{f_{15}} + \frac{1}{f_{\text{new}}}} \quad (3-31)$$

Also, if the cold wind is always in a given direction, the values of U for different exposures (N.E.S.W.) may be made different. Some use values of U with low wind velocity and increase the heat loss on the side exposed to the winds by a given percentage. Such increases of heat losses found from low-velocity values of U are:

TABLE 3-11
EXPOSURE ALLOWANCES

For walls toward prevailing wind.....	15%
For walls on side.....	10%
For corner walls.....	10%

Intermittent Heating. If the buildings are heated intermittently, the heat loss from the space and the radiation requirement for this loss should be increased so that the space may be heated in a reasonable time. If saving of time is desirable the percentage increase given below should be used:

TABLE 3-12
ALLOWANCES FOR INTERMITTENT HEATING

For heating during day and building closed at night, add	10%
For heating during day and building open at night, add	30%
For heating at long intervals (several days), add	50%

Heat Requirements for Air, Ventilation and Leakage. Outside air may be brought into a room by leakage or by ventilating apparatus. The heat required for this air is given by the convection heat formula

$$q_c = Wc_p(t_1 - t_o) \quad (3-4)$$

W represents the pounds of air mixture entering per hour to be heated from a temperature t_o to some temperature t_1 . c_p is the specific heat at constant pressure of the air mixture. Since the air mixture is composed of air and water vapor (determined by the wet- and dry-bulb thermometer reading at some place), the formula above is put into a different form for the amount of dry air in the mixture, W_{da} , and the enthalpies associated with 1 lb of air at the two conditions, h_o and h_i , found from charts of Chapter II, or Table A-2. These give

$$q_c = W_{da}[(h_i - h_o)] \quad (3-32)$$

W_{da} = dry air, pounds per hour;

h_o = enthalpy of 1 lb of dry air and the moisture associated with it;

h_i = enthalpy of 1 lb of dry air and the moisture associated with it.

If the air is measured by volume, V , the quantity W_{da} is found by

$$W_{da} = \frac{V}{v} \quad (3-33)$$

in which V is given in cubic feet per hour, and v is the volume of 1 lb of dry air and its associated moisture given by use of charts of Chapter II, or Table A-2.

In simple or approximate calculations, the air is often considered as dry air, neglecting the small amount of water vapor which it contains, and then the original formula is used in which W is replaced by its equivalent at 70° F and 29.92 in. barometer. From the gas law

$$\begin{aligned} W &= \frac{pV}{RT} = \frac{14.7 \times 144V}{53.35(459.6 + 70)} \\ &= 0.0749V = \frac{V}{13.35} \end{aligned} \quad (3-34)$$

On using the value of specific heat, $c_p = 0.24$ Btu per lb per deg for air, there results:

$$\begin{aligned} q_v &= 0.0749 \times 0.24V(t_i - t_o) \\ &= 0.018V(t_i - t_o) = \frac{V}{55.6} (t_i - t_o) \end{aligned} \quad (3-35)$$

A simpler approximate form is

$$q_v = 0.02(t_i - t_o) = \frac{V}{50} (t_i - t_o) \quad (3-36)$$

The air brought into a space for ventilation is usually at room temperature or above, and the heat is supplied by heaters outside of the room or space to be heated. Such heaters are discussed fully in a later chapter, but the heat needed for the air which enters a room by leakage through the walls or cracks around windows or doors or other openings must be treated as a heat loss from the room and be compensated for by the heat brought into the room by the heating system. Even in indirect heating or in the plenum system where ventilating air is forced into the room by a fan, this leakage occurs and causes an increased heat requirement for the system. The air leakage is dependent entirely upon difference of pressure between that on the outside of the building and that on the inside. The difference may be set up by wind velocity on the outside, low pressure on the inside from

a vacuum ventilating system (suction fan) or by the chimney effect due to the weight of a column of hot air inside of a building differing from the weight of the equal height column of cold air on the outside if such can act for any part of the height of the building.

The amount of air leakage has been the subject of extensive study at a number of universities and at the Experiment Station of the A.S.H.V.E.; * from the results of these investigations, recommended practice is given in the Guide of 1936.

Infiltration. For walls, the leakage with variations of velocity may be given by an approximate formula reduced from the results of Larsen and his associates:

For 13 in. plain brick walls

$$I = 0.041V_w^{1.5} \text{ with } I = 2.39 \text{ at } 15 \text{ mph} \quad (3-37)$$

I = cubic ft of air per hour per square foot of wall;
 V_w = wind velocity in miles per hour.

Plastering reduces the infiltration through $8\frac{1}{2}$ in. walls to 3% of that given by the formula and to 1% for 13 in. walls. Larsen and his associates have shown that a heavy coat of water paint applied to the inside of the plain wall reduces the infiltration to 50% of its value from the formula. Oil paint was not so effective, as two coats cut down the infiltration by 9% and even after three coats the reduction was 28%.

The results of these formulae represent the infiltration through walls laid up in the best manner, and it must be realized that, if care is not exercised to fill mortar joints, even greater infiltration may occur. The effect of plaster applied to the wall is so great that it will eliminate most of the effect of poor work by the brick masons. Plaster applied to furring does not cut down the infiltration quite as much as plaster applied to the masonry. It is to be noted that, in place of the infiltration varying as the velocity which might be assumed for cracks, it varies at a greater rate. The exponent 1.5 was used for simplicity in calculation, although the exponent to give values closer to the experimental results would vary between this and 1.4.

For wooden walls with plaster, such as Fig. 36, the infiltration may be given as

$$I = 0.0017V_w^{1.5} \text{ with } I = 0.150 \text{ at } 15 \text{ mph} \quad (3-38)$$

* See the results of numerous tests reported in the Trans. A.S.H.V.E. for 1916, 1921, 1924, 1927, 1928, 1929, 1930, 1931 and 1932.

The experiments of Larsen and his associates on shingle walls for shelter buildings have been reduced to the same formula by the author

$$I = KV_w^{1.5} \quad (3-39)$$

The values of K are as follows:

$K = 1.60$ for 24 in. shingles placed on 1×6 boards as shingle lath at 11 in. centers;

$K = 0.92$ for 16 in. shingles placed on 1×4 lath at 5 in. centers;

$K = 0.59$ for 24 in. shingles on shi lap;

$K = 0.21$ for 16 in. shingles on shi lap;

$K = 0.002$ for shingles on shi lap with paper between;

$K = 1.22$ for 1×10 in. boards, 11 in. centers with 4 in. battens;

$K = 0.39$ for sheathing matched end and side;

$K = 0.33$ for sheathing matched on side;

$K = 0.62$ for corrugated iron siding.

Window Leakage. The A.S.H.V.E. Guide for 1936 gives the leakage allowances for windows in cubic feet per hour per foot of crack at different velocities and for different window conditions. These are reproduced in Table A-5.

The use of storm sash will cut down the air leakage, but the amount of reduction depends on the tightness of the window to which it is applied. For a loose window the reduction may be 50%, but for a window with weather stripping the reduction cannot be great unless the storm sash is weather-stripped. The storm sash does have a particular value in increasing the temperature of the window glass and so reducing the tendency to condensation of moisture on the room side and resultant frosting in cold weather. In certain regions the double glass formed by the storm sash is necessary for this purpose.

Doors. The Guide recommends that the leakage through doors be computed as that of a poorly fitted window, and if the door is frequently opened, as in a store, three times this value should be used. Because of this excessive leakage, rotating doors are frequently employed at the entrances to office buildings and stores.

The allowances for leakage are usually found for 15 mph, but the Weather Bureau records for the region of the district in which a given structure is to be heated should be examined for the winter months so that the actual velocity and direction can be used. Tabular values given

for the leakage are 80% of the experimental results, as it is assumed that there is some pressure in the building above the atmosphere which offers resistance to flow. If the pressure drop through one wall was 50% of the pressure head caused by the wind, the velocity and discharge might be considered as 70% of that due to full pressure on the one wall. Although there is some justification for this increase to 80% of the test flows, Larsen has shown by experiments on two buildings that the decrease of area of outlet openings, or even the increase of pressure on the opposite side of the wall from that of the room under test, does not decrease the flow of air in proportion to the changes. Even when ventilating air is forced into a room the heat requirements of the leakages must be provided for.

The chimney effect of the hot air within a building may cause a discharge inward for levels below a point at which the outer air and the inner air are in balance, called the neutral zone, and a discharge outward for points above this level. Since this neutral zone is likely to change when windows are opened and leakage occurs through cracks, and moreover, since elevator shafts and stair wells in many modern buildings are cut off by tight doors to prevent such an effect from being set up, this chimney effect, although it may occur, cannot be computed with any degree of assurance.

Having the value of the unit infiltration, the infiltration into rooms of buildings may be computed if these are on the windward side. If the wind is variable and there is more than one side of the room against which the wind may come, then the side having the greater infiltration should be used. The windows and doorways have cracks around sash and doors, and there are also cracks between sash for double-hung or double-casement windows, or for doors when double doors are used. A study of the plans of the building will give the lengths of cracks.

With plastered walls, the leakage is so slight into rooms of the ordinary offices, schools or residences that it may be neglected, using only window leakage. In large auditoriums with little window space the wall leakage may be an appreciable quantity. It should be computed to ascertain its value.

The rooms on the leeward side of the building may need no heat to care for leakage during the prevailing wind, but as the wind may veer it is recommended that even for these rooms a leakage due to a wind of 5 mph be assumed.

To check results of leakage or to allow for it in an approximate way it may be said that the leakage per hour will be equal to one or two times the volume of the room, spoken of as one or two changes per hour.

The leakage having been determined, the amount of heat which must be delivered by the heat supply within the space considered for this infiltration is given by (3-32) or approximately by (3-35) or (3-36).

Carpenter's Rule. The heat loss from any room or space, which is the heat to be added to it, if heat developed by its use is not considered, includes the transmission through the glass or windows and doors, the transmission through the wall and the heat to bring the leakage air to room temperature.

If the average U for glass be taken as 1, that for the wall as 0.25, and if the heat to raise 1 cu ft of air 1 degree F be taken as the approximate value 0.02, the heat required is

$$q = (1G + \frac{1}{4}W + \frac{1}{50}nV)(t_i - t_o)$$

q = Btu required per room per hour;

G = window and door area on sides exposed to atmosphere in square feet;

W = wall area on sides exposed to atmosphere in square feet;

V = volume of room in cubic feet;

n = changes per hour;

t_i = inside temperature in degrees F;

t_o = outside temperature in degrees F.

This, Carpenter's rule, gives a simple and ready method of finding the approximate heat requirement of any space. It should not be used for actual design.

PROBLEMS

1. Find the emission of heat per hour from 1 sq ft of aluminum foil at 30° F to 1 sq ft at 10° F in close proximity. What will be the temperature of a third sheet to deliver this same heat by radiation to the sheet at 30° F? What is the temperature of 1 sq ft of plaster which will radiate the amount of heat per square foot in this problem to foil at 48° F? What would be the temperature drop in $\frac{1}{2}$ in. of gypsum plaster and stucco plaster to transmit this heat per hour per square foot? What is the drop in temperature of an air film transmitting this amount of heat? (still air)

2. Find U for a wall made up of $\frac{1}{2}$ in. of stucco, 3 aluminum foils 0.003 in. thick, 4 air spaces $\frac{3}{8}$ in. thick and $\frac{1}{2}$ in. of gypsum plaster with still air, considering the air spaces to have only the resistance of the air films. Compute the U for this wall using the values of a from Table 3-2 corrected for use of foil. ($t_o - t_i = 70^\circ$ F).

3. If radiant heat transfer is 0.39 Btu per sq ft per hr and the conducted heat is due to a drop of 57.6° F with $U = 0.156$, what is the equivalent U for such a wall? What is the mean k for this if thickness of partition is 2 in.? (still air)

4. Compute the value of U for wall, Fig. 35 F, and check table for $a = 20''$.

5. Compute the value of U for roof, Fig. 39, and check table.

6. Find the U for a wooden wall of 2×4 ($1\frac{3}{4}$ in. \times $3\frac{3}{4}$ in.) studs with $\frac{1}{2}$ in. plaster on wooden lath on inside, $\frac{3}{4}$ in. sheathing, building paper and $\frac{3}{4}$ in. shingles on outside, and 3 sheets of aluminum foil between studs.

7. Find the value of U for the wall D , Fig. 35, if thickness of masonry is increased to 28 in., using value of $U = 0.16$ when thickness is 24 in. What is the value of U if the wind velocity is increased to 30 mph?

8. Find the value of U for the wall A , Fig. 35, if a 1 in. air space is introduced into a 20 in. wall, using the value of $U = 0.21$ for the thickness for 20 in.

9. A room has 60 sq ft of glass and 300 sq ft of wall of $U = 0.20$ on the south side and 100 sq ft of glass and 400 sq ft of the same wall on the west side. What is the sun effect in Btu per hour? What is the amount by which the use of awnings would reduce this sun effect? How much more heat would be removed from the top floor in a similar room if the roof of 1200 sq ft is so constructed that the value of U is 0.10?

10. What temperature would you use for t_o for a building in Philadelphia, Pa.? In Norfolk, Va.? In St. Petersburg, Fla.? What average temperatures would you use for t_i for the walls, ceiling and floors of a room 12 ft high with breathing line at 68° F? For what use would this temperature of room be employed?

11. What leakage occurs from 8 two-leaf metal casement windows, 6 ft high and 4 ft wide over all, with wind at 20 mph? Is this sufficient fresh air for six persons? How much heat would be required to warm this air from 20° F to 68° F?

CHAPTER IV

RADIATORS, VALVES AND HEAT TRANSMISSION FROM RADIATORS

The radiators shown in the figures of Chapter I for hot water or steam heating are of various forms and materials, each of which has peculiar advantages that determine the type selected by the engineer.

A common form of cast-iron radiator, the Corto of the American Radiator Company, Fig. 47, is made by connecting the hubs of several cast-iron sections by close-threaded right and left nipples, as shown in Fig. 48, or by using conical slip nipples or thimbles to join the sections. Through-bolts hold them after they are forced together. The threaded nipples are made with lugs projecting into the interior space so that they may be turned by a plug wrench on the end of a bar. The sections are made of various heights from 20 to 38 in., and in order to give varying amounts of heating surface each height is made of a number of widths, known as three-tube, four-tube, five-tube and six-tube sections. For places beneath low window sills or under seats, the special American Corto window radiators of 14 in., 17 in. and 20 in. heights are manufactured, each of seven tubes. A section, or one loop, as it is called, is $2\frac{1}{2}$ in. from face to face of hub. The dimensions of these sections are shown in Fig. 49.

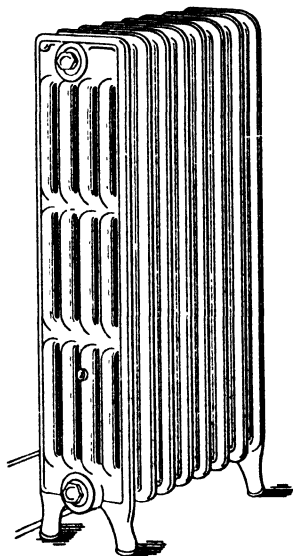


FIG. 47.—American Corto Radiator.

The amounts of heating surface in these radiators in square feet vary with the size as given in the table on page 90:

These areas are given in square feet of equivalent direct radiation (called EDR) determined by test so that each nominal square foot will deliver 240 Btu per hr with steam at 215° F and the room at 70° F.

TABLE 4-1
HEATING SURFACE PER SECTION IN SQUARE FEET
AMERICAN CORTO

No. of Tubes	Height Over All					Width at Floor
	20 in.	23 in.	26 in.	32 in.	38 in.	
Three.....	1.75	2.00	2.33	3.00	3.50	4 $\frac{3}{4}$ in.
Four.....	2.25	2.50	2.75	3.50	4.25	6 $\frac{7}{8}$ in.
Five.....	2.67	3.00	3.50	4.33	5.00	8 $\frac{1}{8}$ in.
Six.....	3.00	3.50	4.00	5.00	6.00	9 $\frac{1}{8}$ in.

Length per section 2 $\frac{1}{2}$ in.

HEATING SURFACE IN SQUARE FEET—AMERICAN CORTO WINDOW RADIATOR

No. of Tubes	Height			Width
	14 in.	17 in.	20 in.	
Seven.....	2.50	3.00	3.67	11 $\frac{1}{2}$ in.

Length per section 2 $\frac{1}{2}$ in.

The Crane Company issues a chart, Table A-6, giving a simple method of selecting the Crane radiators of a given surface to fit the

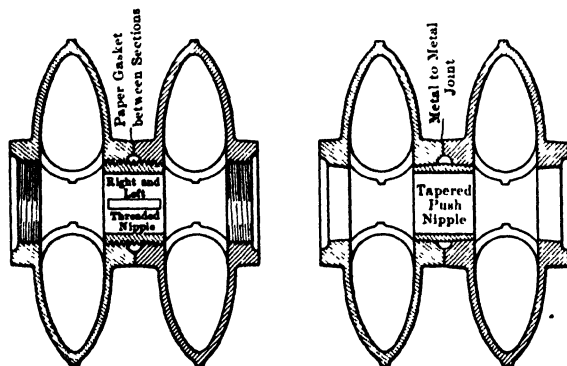
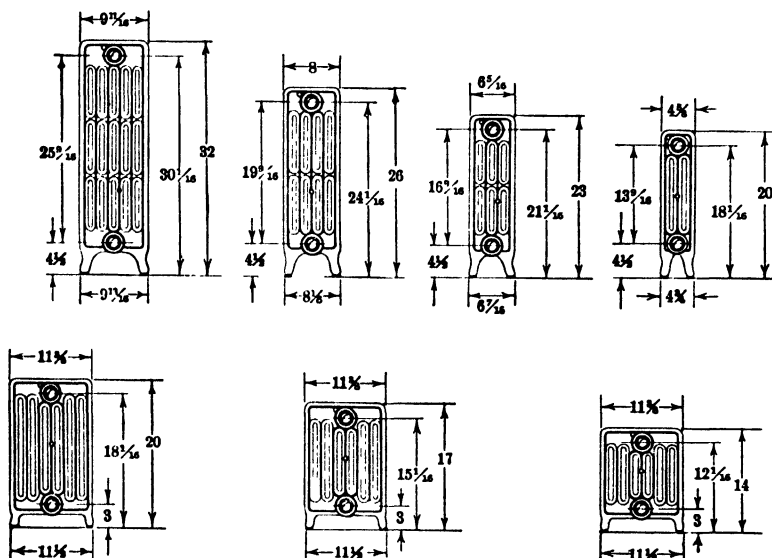


FIG. 48.—Method of Joining Radiator Loops.

dimension of the space available for their placement. Window-sill heights, the heights of window seats or radiator niches fix the heights which may be used; depths vary with the space available. To provide space for the radiator connections and valves, the length of the radiator should be at least 6 in. less than the length of the space available for it. As the Crane dimensions are practically the same as the tube radi-



Courtesy of American Radiator Co.

FIG. 49.—Dimensions of Corto Radiators.

ators of other manufacturers, this chart may be used for the selection of any tube units. Similar charts may be constructed for the radiators of other manufacturers, although their publications give total surface and space requirements of radiators made up of different numbers of sections.

The Arco type of the American Radiator Company contains fewer tubes than the Corto radiators, but of greater width. The type is used by persons who desire this form of radiator for reasons of personal preference. This type is built of three heights: 19 in., 22 in. and 25 in., and of two widths, three-tube and four-tube. The sections are $1\frac{1}{2}$ in. thick from face to face of hubs. The surface per section of the three-tube units is respectively, 1.1 sq ft, 1.3 sq ft and 1.5 sq ft; that for the four-tube units is respectively, 1.4 sq ft, 1.6 sq ft and 1.8 sq ft.

Hospital radiators are made with sections spaced at greater distances and without any beads or flutings in which dirt may collect. In

general they are carried on wall brackets, Fig. 50, to eliminate the feet which rest on the floor. All parts of the surface must be available for cleaning. The Corto sections are finished for hospital use by increasing the distance from hub to hub to 3 in. and by omitting beads. They are built in three- and five-tube widths and have the tabulated heating surface per section. The length over all for a given number of sections is 20% greater than that of the same number of sections of standard Corto radiators.

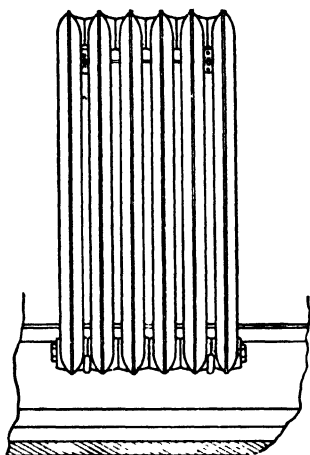
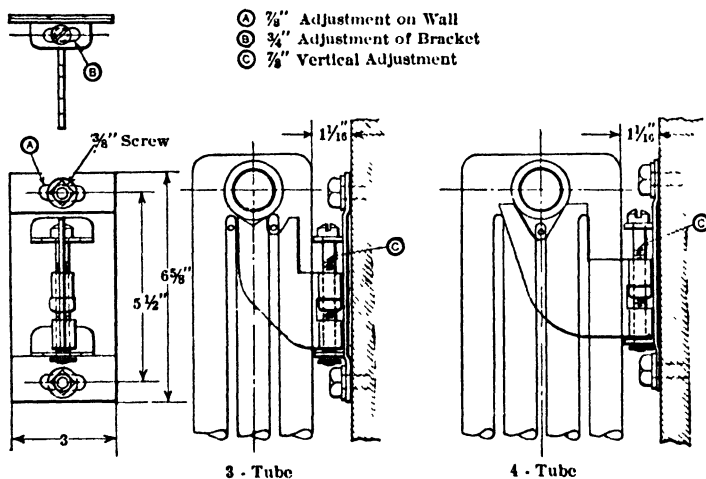


FIG. 50.—Hot Water Radiator on Brackets.

When radiator sections are connected only at the lower end, the type is called the steam radiator type; that in which the connections are made at top and bottom is called the hot water radiator type. The better distribution of steam in the radiator which comes from the double connection has the result that this

hot water radiator type is the prevalent one for steam as well as for water. This type is clearly indicated in Fig. 50, which shows a radi-



3 - Tube

4 - Tube

Courtesy of American Radiator Co.

FIG. 51.—Radiator Bracket.

ator supported on wall brackets, Fig. 51, so as to give a clear floor line, free from radiator feet.

Radiator sections have been made of sheet copper and sheet iron, like the early Kinnear and Pressweld radiators, which were very light in weight.

Special Radiators. The radiator sections may be combined to fit into square corners or arranged in curves. They may be joined to fill the jib panel of a stairway and even combined to form a dish-warming cabinet. All these special combinations must be fabricated at an additional cost.

If it is necessary to raise the radiator to accommodate piping or some change in building construction, a form of radiator pedestal is placed beneath the feet.

Wall Radiators. Wall radiators are used when the space for radiator thickness is limited. These are shown in Fig. 52.

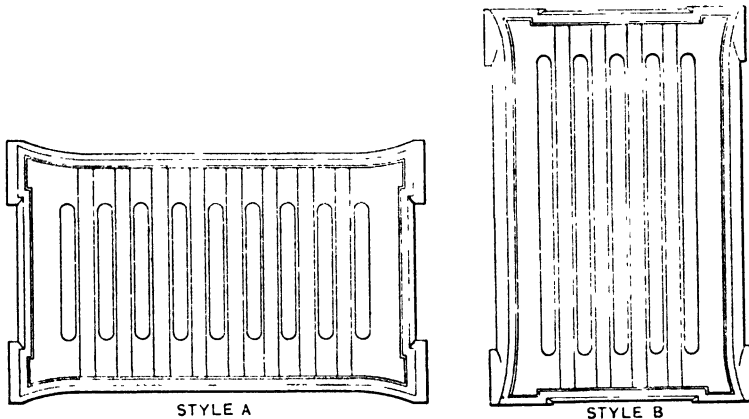


FIG. 52.—Peerless Wall Radiators.

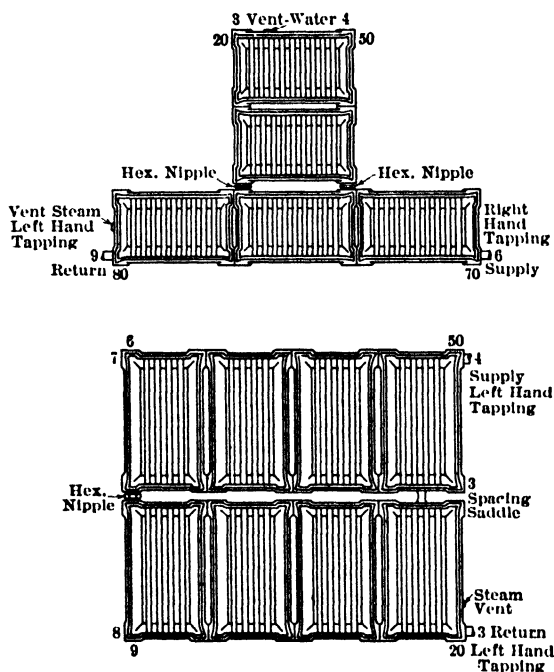
The Peerless wall radiators are made of the dimensions of Table 4-2. These are over-all dimensions, and when the sections are connected by nipples they come close together, section to section.

TABLE 4-2
PEERLESS WALL RADIATOR DATA

Section No.	Length	Width	Thickness	Area
5A.....	16 $\frac{5}{8}$ in.	13 $\frac{5}{8}$ in.	2 $\frac{7}{8}$ in.	5 sq ft
7A and 7B.....	21 $\frac{7}{8}$ "	13 $\frac{5}{8}$ "	2 $\frac{1}{2}$ "	7 " "
9A and 9B.....	29 $\frac{1}{8}$ "	13 $\frac{5}{8}$ "	2 $\frac{1}{8}$ "	9 " "

Area based on 300 Btu per sq ft per hr at 215°-70°, Type A. (270 Btu. Type B).

These castings are tapped on the corners so that a number of them may be connected by right and left close nipples with inner lugs, or by nipples with outside hexagonal projections at center to form one radiator of desired surface to fit the available wall space, as shown in Fig. 53. In this figure the numbers 2-9 represent regular tapplings of $1\frac{1}{2}$ in. while the numbers 20 to 90 represent special tapping if desired. If hexagonal nipples are used, $1\frac{1}{8}$ in. is added to length. The use of hexagonal nipples



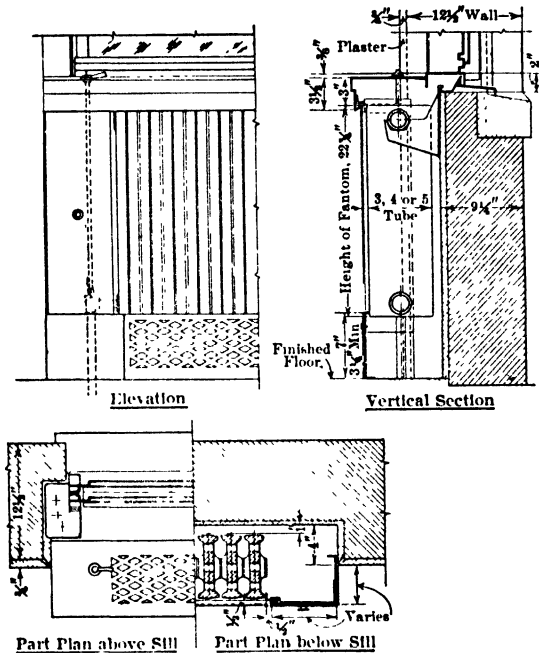
Courtesy of American Radiator Co

FIG. 53.—Peerless Wall Radiator Combinations.

necessitates a spacing saddle at the opposite end of the radiator for support. These units are carried by Arco adjustable brackets on the walls.

Flue Radiators. Radiator sections are sometimes made with projecting ribs on their faces so that when connected by nipples or thimbles a series of flues is formed. Such radiators are known as flue radiators; Fig. 54 illustrates the Phantom radiator of the American Radiator Company. The flues are seen in the plan view below the sill. The radiator is supported on brackets in a window recess and is covered by the perforated Campbell metal window sill. The grilles at base and top per-

mit air circulation for the convection of heat, while the face of the radiator delivers radiant energy. The spaces at each end of the radiator for connecting pipes are faced with metal which, when painted to match the color of the radiator, give an attractive finish. With sufficient wall thickness the window recess may be deep enough to give a face flush with the wall. Otherwise the sill, face and ends make a radiator enclosure.



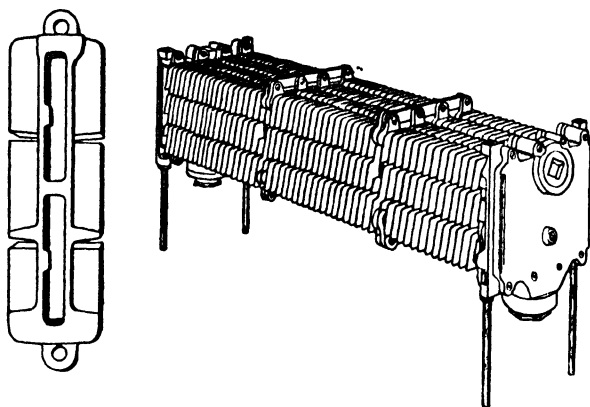
Courtesy of American Radiator Co.

FIG. 54.-- Phantom Radiator.

Concealed Radiators. On account of the desire of architects and owners to eliminate the open radiator from the room, many conceal the radiator by a cover or by placing it in a recess with a panel or grille cover. As the heat transfer is reduced by this covering, the attempt was made to permit a large circulation of air with an increase in the surface in contact with it by the addition of fins to the flat radiator surface. The concealment of the radiator eliminates any consideration of its appearance, and this increase of surface by fins reduces the over-all dimensions of the radiator as well as its weight. A number of radiators have been built to meet these new requirements.

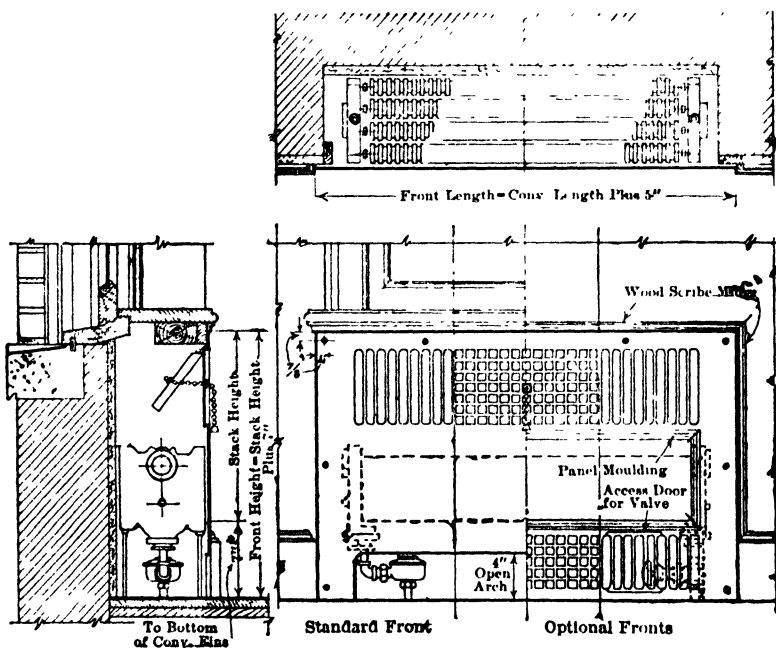
Fig. 55 illustrates one of these concealed radiators, the Arco con-

vector of the American Radiator Company, built up of any number of cast finned sections arranged to be attached to cast headers at each end.



Courtesy of American Radiator Co.

FIG. 55.—Arco Convector.



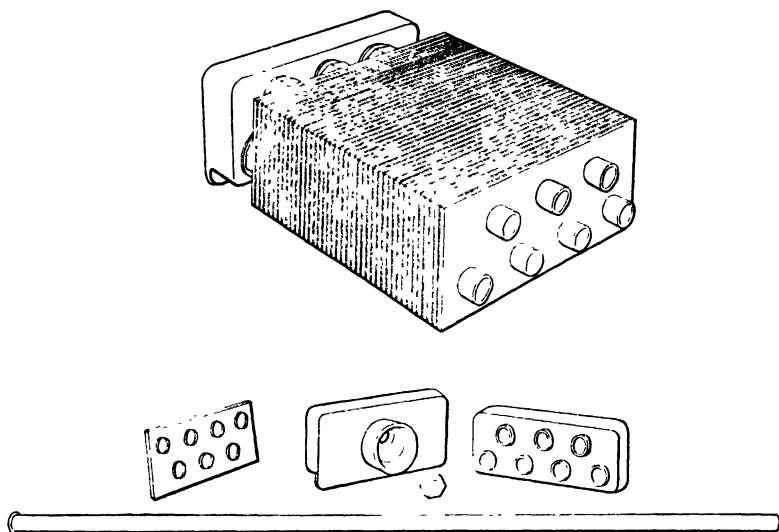
Courtesy of American Radiator Co.

FIG. 56.—Enclosed Arco Convector.

The headers vary in widths so that from two to five sections may be used. By varying these and the number of units in length, many heat

demands may be met. The application of this to the window recess in a masonry wall is shown in Fig. 56. This convector has four tubes in width. The grilles, valves, piping, sill and panel are shown as well as the insulating material on the face of the wall.

Fig. 57 shows the copper fin-plate radiator, built by John J. Nesbitt, Inc., for a unit heater. In this type seven $\frac{3}{4}$ -in. copper pipes are connected to iron headers at each end, one of which is connected to the steam supply and the other, at a lower level, to the return. Copper plates 3 in. \times 8 in. are forced over these pipes, making a tight fit for



Courtesy of John J. Nesbitt, Inc.

FIG. 57.—Nesbitt Extended Fin Plate Radiator.

good heat conduction. The plates are maintained at a proper distance by the small flanges turned up at the pipe holes of the plates and by the right-angle flanges formed at the edges. These plates are placed at intervals of $\frac{1}{8}$ in., $\frac{3}{16}$ in. or $\frac{1}{4}$ in., so that there is enormous heating surface for a given space. This radiator is equivalent to a large heating surface and is used to heat air delivered against it by a fan.

Pipe Coil Radiators. Pipe coil radiators, called pipe coils, are very satisfactory. They are made by joining pipes together by return bends, Fig. 58, or by branch tees or manifolds, Fig. 59. It must always be remembered that some of the pipes are likely to heat before all are heated, so that such a construction as shown in Fig. 60 is improper. The expansion of the top pipe before the other pipes are heated may cause the rupture of the branch tees. The arrangements shown in Figs.

58 and 59 permit the expansion of any individual pipe without straining the system. In all coil construction any one pipe may expand more than another.

Fig. 61 is a coil used at the corner of a room. In this form of coil,

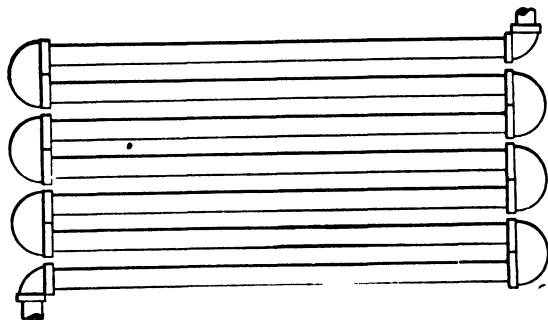


FIG. 58.—Pipe Coil Radiator with Return Bends.

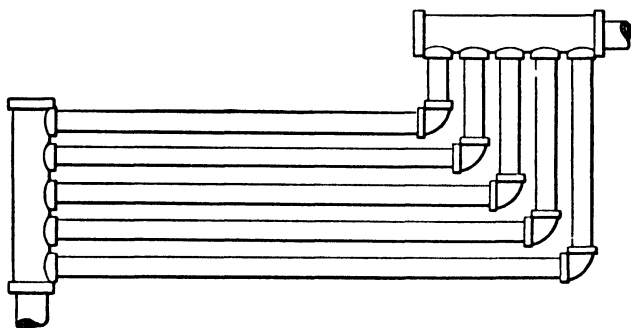


FIG. 59.—Pipe Coil Radiator with Manifolds or Branch Tees.

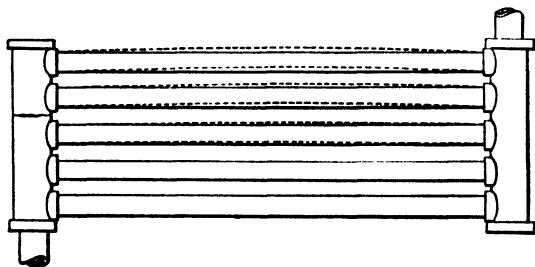


FIG. 60.—Improper Form of Pipe Coil Radiator.

expansion may occur in any pipe, and for this reason this particular arrangement and that of Fig. 59 are used very often.

For the purpose of giving data to be used for pipe coil construction as well as the design of pipe lines, Table A-7 of the Appendix is included

for the dimensions of standard wrought-iron and steel pipes. The pipes used should always be of full weight, and after cutting to length they should be reamed out.

Finned Pipes or Tubes. A method of increasing the heat transfer from pipes when used for air heating, as in the Nesbitt heating unit, is the use of fins. In one type, known as Aerofin tubing, Fig. 62, a seamless copper pipe is equipped with a thin helical copper ribbon, crimped around the outer surface. "Aerofin" tubing increases the primary tube area by 400% in certain sizes, although different widths of ribbon are used as well as different numbers of turns per inch to meet the demands of practice. The crimping increases the length of surface of the fin in contact with the tube, and this edge is brought into better thermal contact by solder. The crimping produces turbulence in air flow across the tube and fin, reducing the thickness of the air film and increasing the rate of heat transfer.

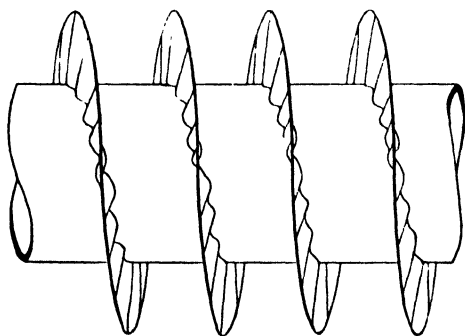


FIG. 62.—Aerofin Tube and Fin.

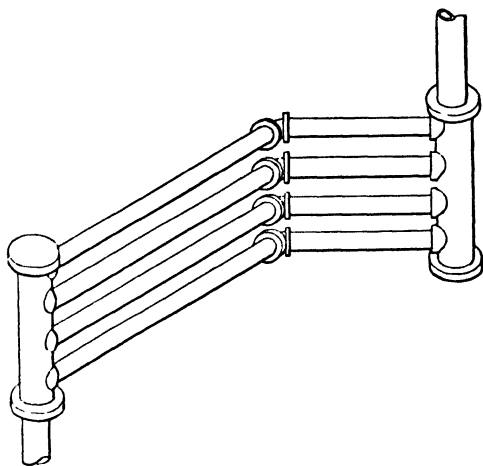
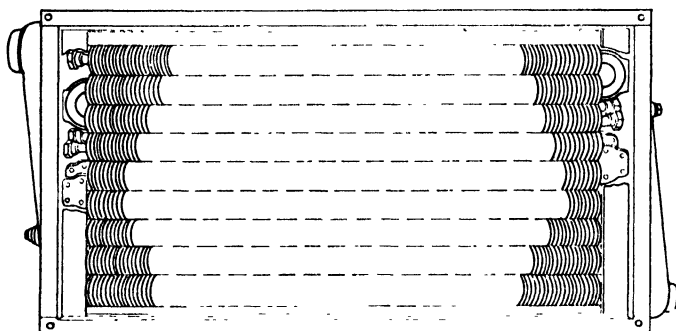


FIG. 61.—Corner Pipe Coil.

Three arrangements of these finned pipes are built into units in a metal frame with headers at each end. The universal type (to 200% pressure), Fig. 63, is arranged with three pass coils between the headers. The flexitube type, Fig. 64, consists of 18 single pass tubes between the

headers. This is built for pressures up to 200%. The high-pressure type is used for pressures as high as 350%. The flexitube type may be placed horizontally or vertically because of the shape of the tubes and because of their single length. The tubes are brazed to the brass plate

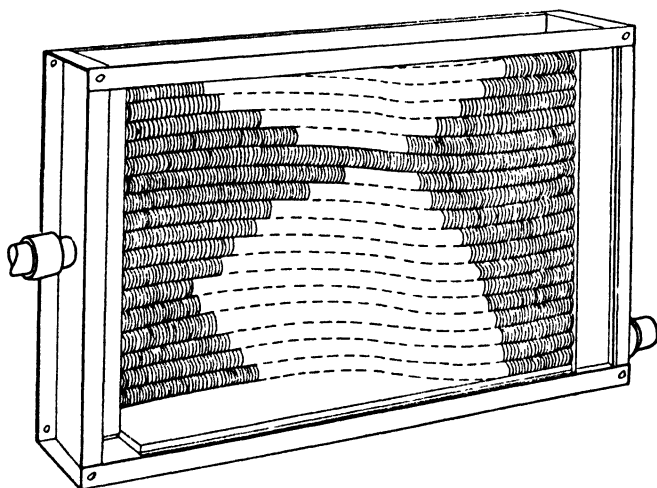
headers at each end and the ends of the tubes are flanged over to form small inlets so as to limit and equalize the flow of steam to all tubes. The main advantage of this form of heating surface over that to be



Courtesy of Aero-fin Corporation

FIG. 63.—Universal Aero-fin Unit.

explained later is the great reduction in weight and depth of heater or length of air pass. Should the desired area required for full air inlet be such that no single standard size can be found, or the amount required

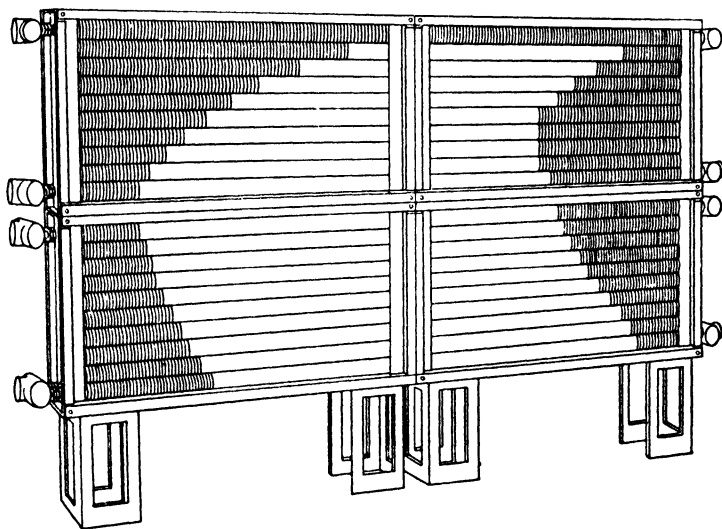


Courtesy of Aero-fin Corporation

FIG. 64.—Flexitube Aero-fin Unit.

is greater than the maximum standard size, an arrangement such as that shown with four units in parallel, Fig. 65, may be used. The units shown here are of the high-pressure type in which the coil is ten tubes

high with bends at the ends to form a continuous coil. The temperature desired for the outlet air at the rear of this unit fixes the number of units in series, and if in Fig. 65 two are required in depth, a second set of four units is placed behind these. This is called a series arrangement for the air flow to give the desired outlet temperature. The resulting temperature as well as the heat transfer from these units at different velocities and entering temperatures have been determined by the manufacturer and are reported in their data books.



Courtesy of Aero-fin Corporation

FIG. 65.—High Pressure Aero-fin Units in Parallel.

For indirect heating through air delivered at a higher temperature than that of the room, the extended surface and fan, the convector or the Nesbitt type of heating unit is used, but for many years previous to the introduction of copper fin heaters, closely placed pipes attached to cast-iron bases of special forms were used. In the forms of the Sturtevant Company and the Buffalo Forge Company, Fig. 66, four rows of 1 in. pipe were connected to the top of the supply and discharge chambers of a cast-iron base, the separate pipe rows being staggered to produce turbulence in the air flow as it passed against the faces of these pipes. Certain pipes are arranged with right and left threads at the ends to permit the assembly of the unit. Steam is supplied to the chamber on the right-hand side of the base, and the condensation which reaches the left side is discharged through the chamber at the bottom. There may be some condensation from the vertical pipes on the right

which drops back against the steam flow. This is cared for by the small drip hole at the left end of the supply chamber. Resulting temperatures and heat transfer data of these units are furnished by the manufacturer.

Cast-iron radiators for indirect heating are made by the American Radiator Company. These are known as Vento cast-iron heaters and are shown in Fig. 67. They are made up of hollow cast-iron sections, made like long boxes of rectangular cross-section with occasional ties or cast stays between the two faces. On the faces of these boxes hollow

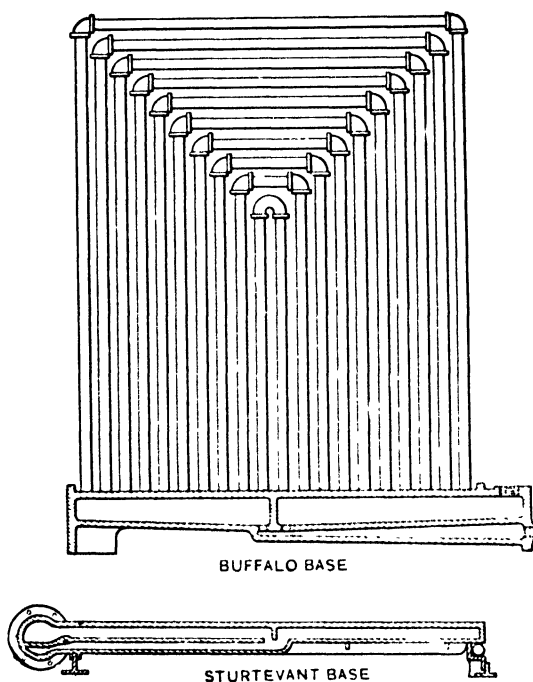


Fig. 66.—Coil Heater Section.

diamond-shaped projections are formed, the alternate lines of which are staggered so as to break up the air flow. A number of sections of the heater are connected to form a stack by right and left hexagonal nipples, Fig. 68, screwed into them by means of a wrench. The sections are brought close together so that the diamond-shaped projections almost touch when the nipples are screwed tight, the distance apart being regulated by the length of nipple. There are four standard lengths of nipples to vary the area between sections for air flow. The largest number of sections made into one stack in regular practice is twenty-four, and when the quantity of air requires more frontal area than that

given by this number, several stacks are placed in parallel for air flow in tiers, side by side, or in a combination of these, as was shown in Fig. 65. Here also the number of stacks placed back of each other, or in series for air flow, is fixed by the required outlet temperature, using the manufacturers' data sheets to determine this.

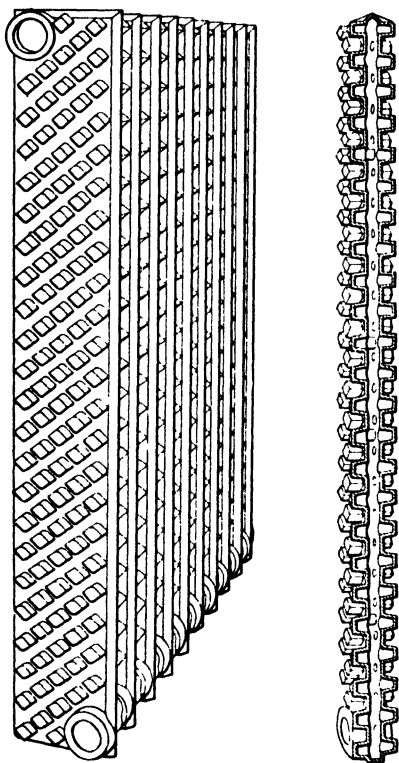


FIG. 67.—Vento Heater Stack of Ten Sections.

To fit various conditions, the Vento sections are made of five different lengths, as given in Table A-13.

For indirect heating with natural circulation, Fig. 2, the American Radiator Company manufactures a cast-iron radiator, Fig. 69, the Perfection Pin Radiator. A cast-iron box of rectangular sections has a series

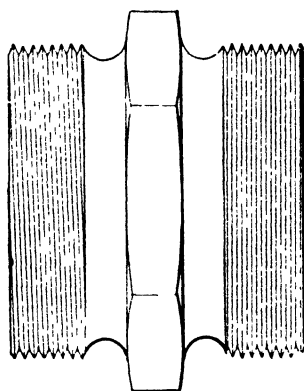


FIG. 68.—Right and Left Vento Nipple.

of pin projections from the large flat surfaces. The steam is supplied at the upper corner, and the drip is taken from the lower one. Two sizes are built: one, the standard, $36\frac{1}{4}$ in. \times $9\frac{1}{8}$ in. \times $2\frac{1}{8}$ in. over all, contains 10 sq ft of heating surface; the other, extra large, has a height of 14 in. and contains 15 sq ft. The pins project $\frac{1}{2}$ in., and the standard spacing of the units is $2\frac{3}{4}$ in. for the first and $2\frac{7}{8}$ in. for the second, giving $1\frac{1}{8}$ in. and $1\frac{1}{4}$ in. between flat surfaces. A still larger unit known as the Sanitary School Pin Radiator is $36\frac{1}{8}$ in. \times $15\frac{1}{2}$ in. \times $3\frac{1}{8}$ in. over all and contains 20 sq ft of heating surface. In this the

pins project 1 in. and the sections are centered at 4 in. as a standard, giving $2\frac{1}{2}$ in. between flat surfaces.

Panel Warming.* About 1909, A. H. Barker of England proposed the use of panels at 200°F formed in the ceiling or walls of buildings to supply heat by radiation to the occupants of the rooms. In Liverpool, a building of 1000 rooms using panel heating has been operating successfully for almost thirty years. Many buildings in Great Britain and a few in the United States have been so equipped. The general plan is to balance the heat conducted outward from the walls at lower temperatures than the rooms, because of the drop in temperature in the air film and the conduction of these walls, against the heat radiated from the panels and the occupants of the rooms. For comfort the air conditions around the occupant must be such that bodily heat is released at a

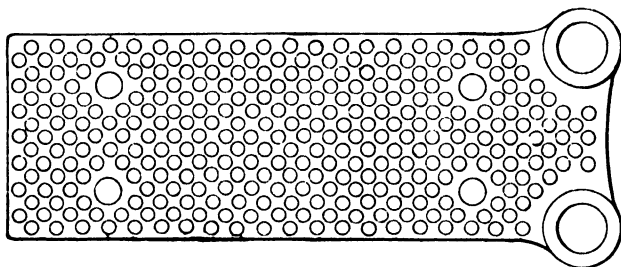


FIG. 69.—Perfection Pin Radiator.

proper rate so that there will not be a feeling of heat or cold. The lower heat loss from the walls with lower temperature makes this system less costly to operate than systems using higher temperatures.

Overhead Direct Radiators. The return piping of a building may be at such an elevation that the only place available for radiators in the basement is the ceiling. Such positions are inadvisable as much of the heat remains at the ceiling and the high rate of radiation from steam radiators to occupants below is uncomfortable. They must be used, however, in certain installations.

Radiator Valves. The Jenkins quick-opening low-bonnet valve, Fig. 70, placed on a radiator to control the supply and exhaust of steam or water is built with a metal body *E* in which there is a partition *F* between the entrance and exit which contains a circular opening closed by a valve disc *B* attached to a spindle *G*. When this spindle is rotated by turning handwheel *M*, it is moved axially by means of its screw

* Panel Warming, by L. J. Fowler, Trans. A.S.H.V.E., Vol. 36, 1930.

Room Warming by Radiation, by A. H. Barker, Heating, Piping and Air Conditioning, March, 1932.

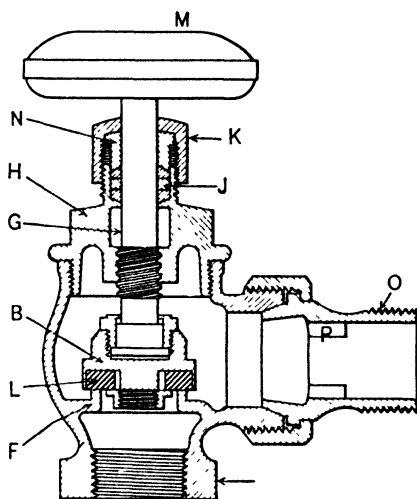
thread turning in a thread in the bonnet *H*. The spindle *G* passes through packing *J* in a stuffing box on the bonnet, and the packing is tightened against the spindle by the gland *N* and the cap *K*. A Jenkins renewable disc *L* attached to the valve disc on the spindle can be renewed when worn by removing a small nut on the bottom of the metal valve after withdrawing the bonnet.

The valves ordinarily used on steam lines are connected with inlet and outlet in one line (straightway), but for radiator valves these are built, as shown, as angle valves, and to eliminate gaskets the outlets have ground joint unions between the valve bodies and the nipples *O*. These latter fit into the threaded openings of the radiators. The nipples have projecting lugs, *P*, on the inside so that they can be screwed into the radiator tap-pings by means of a plug wrench.

As space requirement is an important item in placing radiators, this design is of great value.

If this valve is used for hot water a small hole is made through the partition *F* to permit a slight circulation when the valve is closed and prevent freezing in cold weather. The stem, or spindle, of the valve, Fig. 70, requires packing which must be renewed and adjusted at intervals. To eliminate the need of this, the American Radiator Company has adapted the corrugated metallic bellows or sylphon in the construction of the sylphon packless radiator valve, Fig. 71. In this valve the bellows *A* is attached to the valve disc *B* and to the bonnet at the top. This encloses the spindle in a space which is free from steam. The sylphon is made of steam brass and will stand repeated extensions without rupture.

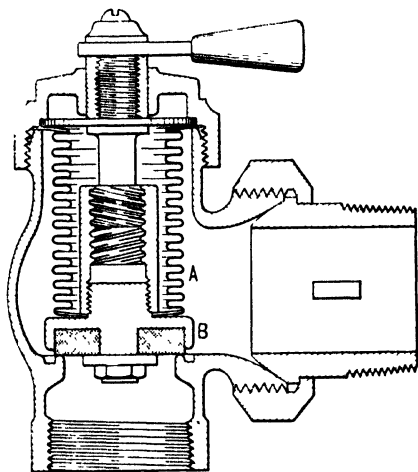
For hot-water radiators, angle valves are sometimes built with plug cocks, or their equivalent, in which a quarter turn of the spindle opens the valve its full extent. The *QO* water radiator valve (quick opening) is shown in Fig. 72. The valve is so made that the diagonal disc comes in contact with the body of the valve at top, bottom and sides, and forms a partition between the outlet and the inlet of the valve, or con-



Courtesy of Jenkins Bros

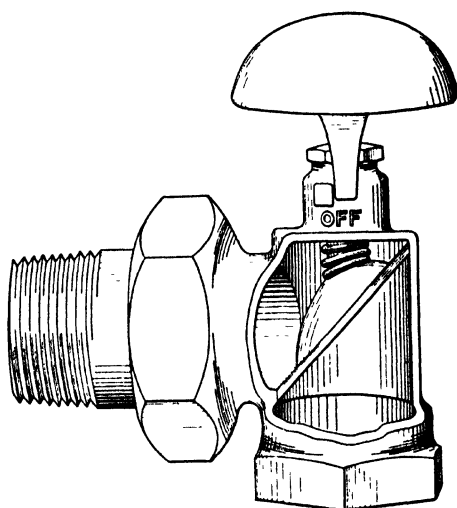
FIG. 70.—Jenkins Radiator Valve.

nects them. Wear is provided for by tapering the parts, and a spring at the top presses against a washer around the spindle of the valve and forces the disc against its seat.



Courtesy of American Radiator Co

FIG. 71.—Arco Packless Radiator Valve.



Courtesy of American Radiator Co

FIG. 72.—Arco Q. O. Water Radiator Valve.

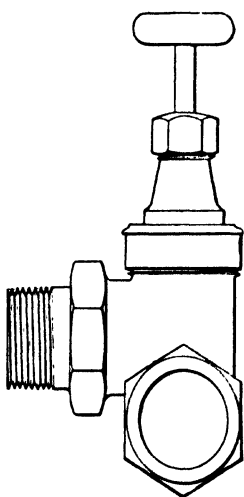


FIG. 73.—Offset Corner Valve.

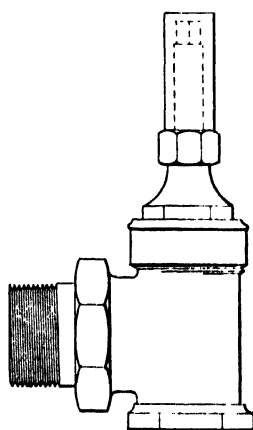


FIG. 74.—Lock and Shield Valve.

An offset corner valve, Fig. 73, of either right- or left-hand construction is necessary at times on account of the position of the piping. It reduces the number of elbows and nipples needed and shortens the con-

nection. It is called offset because the inlet and outlet openings are at different levels. Valves with inlet and outlets in one line, called globe valves or offset globe valves, are made with the ground joint unions to meet peculiar conditions.

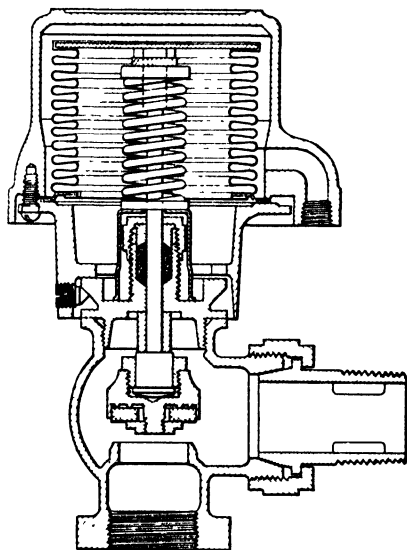
Radiator valves are sometimes regulated by a key when the control of the valve by the room occupant is to be made impossible. In such valves, Fig. 74, the spindle is within a shield into which a key or socket wrench may fit.

Radiator valves are operated by air in certain systems of temperature control, and Fig. 75 shows a sectional view of a Johnson angle radiator valve with its operating motor on top. Air is introduced outside of the syphon bellows to cause the valve to close against the pressure of an opening spring placed around the valve spindle. The withdrawal of the air permits the pressure from the spring to open the valve.

There are systems of temperature control in which the disc of each thermostatic radiator valve is attached by a lever to a thermostatic motor element in a casing at one side of the valve. This element forces the valve to its closed position when the temperature of the room adjacent to the radiator reaches the point for which the thermostatic element is set.

The simplicity of operation of the quarter-turn hot-water valve has led to the design of a number of steam valves in which the valve can be opened its maximum amount by a half or quarter turn of the spindle through the handle attached to its outer end. The valve is raised by a slotted cam or screw thread of steep pitch angle, and in many cases the cam can be arranged to give a slow motion for most of its handle movement, slowly increasing the amount of steam as far as a point near the end of the movement, from which location a small turning of the handle fully opens the valve.

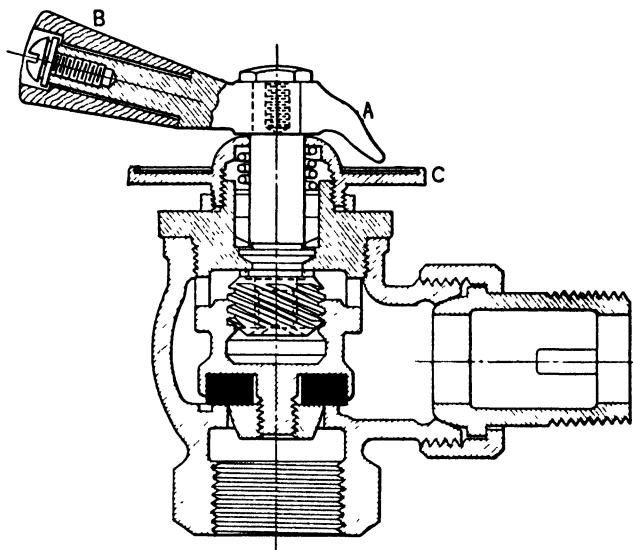
One such valve is the modulation valve of Barnes and Jones, Fig. 76. The size of the valve is so selected or adjusted with reference to the



Courtesy of Johnson Service Co.

FIG. 75.—Johnson Air Motor Radiator Valve.

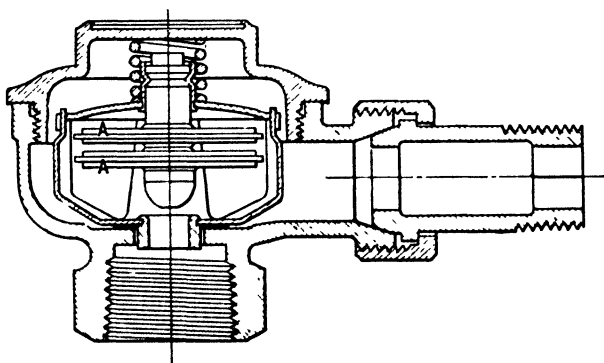
radiator surface that when the pointer *A*, opposite the handle *B*, is at a point marked $\frac{1}{4}$ on the dial *C*, beneath it, the valve has been raised sufficiently to admit the quantity of steam needed to heat one-fourth



Courtesy of Barnes and Jones, Inc.

FIG. 76.—B and J Modulation Radiator Valve.

of the radiator. This valve is connected to a hot-water type of radiator, Fig. 78, so that the steam is shot across the full length of the radiator,



Courtesy of Barnes and Jones, Inc.

FIG. 77.—B and L Return Radiator Valve.

heating all sections to about a fourth of the height from the top. Since the heat from the amount of steam thus admitted is transmitted to the

room by this extent of radiating surface, the steam is condensed by it and the rest of the radiator remains cold.

The outlets of radiators have been controlled by hand valves, but then it is necessary to close both supply and return valves to cut off a radiator, and, moreover, if too much steam is admitted to any one radiator, pressure will build up in the return lines through the open outlet valve. Modern practice has demanded a valve which will prevent the outlet of steam while permitting the discharge of condensate and air, and so various return valves in the form of steam traps have been devised.

The return valve (outlet trap) of Barnes and Jones, Fig. 77, is in reality a trap which will not allow steam to pass. It consists of two diaphragms or expandable units of phosphor bronze, *A*, which contain a volatile fluid such as gasoline. When these are cold the collapse of the diaphragms raises the valve, the construction being such that the valve disc or plug is lifted from its seat at this time. When steam reaches the valve, the heat volatilizes part of the liquid in *A* and produces sufficient pressure to push the valve stem down against the seat, closing off the valve. The spring at the top permits a definite seating pressure.

The two valves are shown in their position on a radiator in Fig. 78, and with valves of this or equivalent makes, air valves are unnecessary, as air is driven out from the bottom of the radiator in advance of the entering steam, the return valve remaining open until steam reaches the diaphragm.

Air Valves. Where steam is admitted and condensate removed from the bottom of a radiator the control valves will not relieve the air. When a single pipe cares for both supply and return (one-pipe system) using only one radiator valve, air may be retained in the radiator. An automatic air valve must be used on each radiator in this system as a collection of air from the steam or water interferes with the proper heating of the radiator and causes noise. These air valves are hand-controlled and automatic. The hand-controlled compression air valves

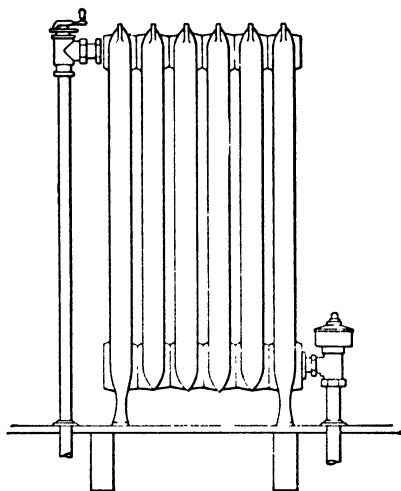


FIG. 78.—Hot Water Radiator with Modulating Valve and Trap.

are small conical angle valves, Fig. 79, and are commonly employed on the radiators used in hot-water heating.

The expansion air valve, Fig. 80, is used in single-pipe heating systems in which the outlets of all air valves are connected to a discharge air-pipe system maintained at a pressure much below that of the steam supply. In this valve the core or spindle is made of a material which has a greater coefficient of expansion than that of the body of the valve. These coefficients are so different that when the plug core or spindle has

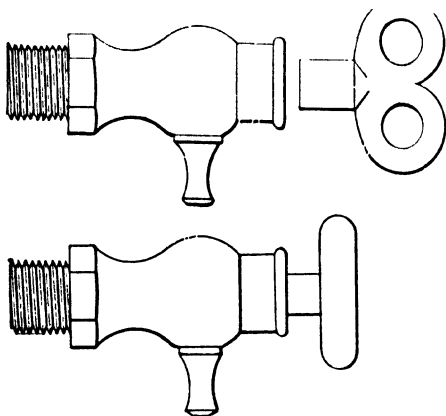


FIG. 79.—Compression Air Valves.

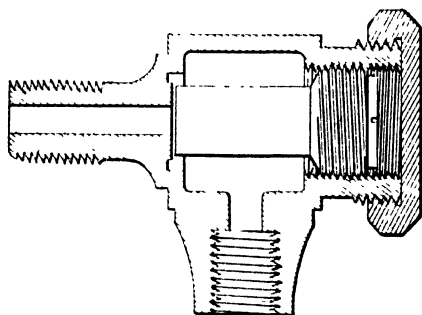


FIG. 80.—Expansion Air Valve.

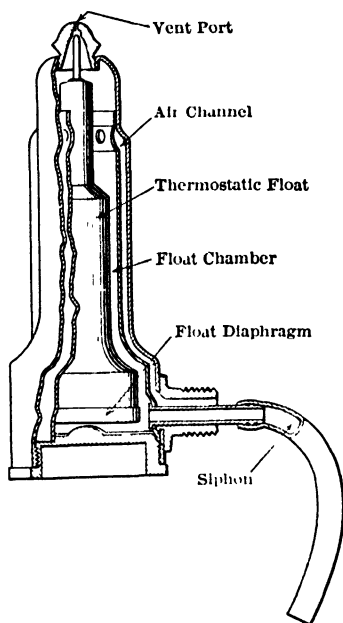
been adjusted by a screw driver the valve will be open when cold but will close when steam reaches the outlet. When the valve is open, the vacuum in the vent line removes air from the radiator. The valves may also discharge freely into the room to be heated, although this is objectionable on account of odors and water which may issue from the valve.

The air outlet valve of the Hoffman Specialty Company, Fig. 81, operates to allow air to discharge through the upper outlet when cold. The brass float in the center contains a volatile fluid which, on being heated, presses on a flexible metal diaphragm at the bottom of the float and forces the seating pin against the seat as the diaphragm presses on the supporting post. The float will act also to prevent water from being forced out of the air valve. The bent tube allows condensed steam to flow into the radiator from the valve against any inward flow of steam or air above the tube.

Heat Transmission from Radiators. Radiators described in this chapter receive heat from steam or hot water within them, although similar action will take place from other fluids. Even in the warm-air

furnace of Fig. 1, the heating surface of the radiator and its connected parts receives heat from a heated fluid, the products of combustion from the fire. In all of these a hot fluid brings heat to the surface by convection from some source of heat energy, and the surface of the radiator must receive heat from this hot fluid by heat transfer through a film of the hot substance and by radiation from any molecules of the fluid which are above the temperature of the radiator surface. It then delivers this heat to the space or material on the other side. The general formulae of radiation, conduction, convection and heat transfer of Chapter III apply here, and it remains to discuss the particular features of these for the radiators used. The radiator surfaces are relatively thin metal conductors, so that the thermal resistance of the material is particularly small and the principal resistances to heat flow reside in the films on the two sides of the surface. This heat transfer differs from the transfer through the walls of Chapter III, in which films did not offer as much resistance as the other parts of the partition. It must be realized also that the side which has the greater thermal resistance of film is the side which controls the heat flow. In general, in steam and hot-water radiators, the side in contact with the air to be heated is the side that limits the heat flow, and as a result the surface of the radiator approaches the temperature of the steam or water. In hot-air furnaces there is much resistance to heat flow in the films on each side and the metal takes an intermediate temperature; in boilers and hot-water heaters the metal walls take a temperature nearer that of the water because of the greater thermal resistance of the film of the products of combustion in comparison with that of the water film and the thin metal wall.

The unit thermal conductance for heat flow through a film, k/L , has been found to depend on the friction factor, f , of the hydraulic flow of the fluids, the specific heat of the substance and the flow of substance in pounds per hour per square foot of area of passage, called mass velocity.



Courtesy of Hoffman Specialty Co.

FIG. 81.—Radiator Air Valve.

The value of this quantity f would fix the heat flow if the temperature drop in this film were known. However, instead of using a general method of determining the unit thermal conductance of the film, and from it the heat transfer, experiment has been resorted to for the determination of U for each type of radiator and its variation with changes of conditions. In these experiments the amount of heating medium per hour has been measured and the enthalpy, h , at entrance and exit determined; then the heat q , per hour is given by

$$q_t = 3600 W(h_{\text{ent}} - h_{\text{ex}}) = UA(t_s - t_r) \quad (4-1)$$

In this equation all quantities except U may be measured experimentally, and from it the value of U is found and tabulated.

In (4-1), t_s is the temperature of the heating medium, which is assumed to be steam, for which the condensate W has been measured in pounds per second. t_r is the room temperature. When steam is used, t_s is constant over the whole surface, and t_r is practically the same. To find the mean value of $(t_s - t_r)$ when both t_s and t_r vary requires the consideration of the heat transfer of differential areas and the integration of these. The expression for q , then becomes

$$\begin{aligned} q_t &= 3600 W_{\text{hot}}(h_{\text{ent}} - h_{\text{ex}}) = 3600 W_{\text{cold}}(h'_{\text{ex}} - h'_{\text{ent}}) \\ &= \int U_x \Delta t_x dA \end{aligned} \quad (4-2)$$

$\Delta t_x = t_{hx} - t_{ox}$ (temperature difference at point x)

$$q_t = (U \text{ for mean } \Delta T)A(\text{mean } \Delta t) \quad (4-3)$$

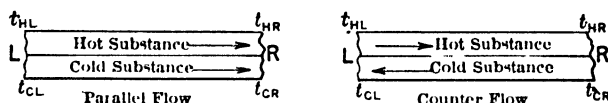


FIG. 82.—Parallel and Counter Flow of Fluids in Heat Transfer.

In the determination of mean Δt this equation is used to define mean Δt .

The substances on the two sides of a surface may flow in two different ways: (a) parallel flow, in which the materials on the two sides are moving in the same direction; and (b) counter flow, in which the fluids are moving in opposite directions. These are shown in Fig. 82.

Assuming the variation of A and t_h , the temperature of the hot substance, to be positive from right to left

$$U_x dA(t_{hx} - t_{cx}) = + 3600 W_h C_h dt_h = \pm 3600 W_c C_c dt_c \quad (4-4)$$

The upper sign refers to counter flow and the lower to parallel flow.

W_h = pounds of hot substance per second;

C_h = specific heat of hot substance;

t_{hx} = temperature hot substance at x ;

W_c = pound of cold substance per second;

C_c = specific heat of cold substance;

t_{cx} = temperature of cold substance at x .

Integrating from X to L

$$\int_x^L U_x \Delta t_x dA = 3600 W_h C_h (t_{hL} - t_{hx}) = \pm 3600 W_c C_c (t_{cL} - t_{cx}) \quad (4-5)$$

$$\frac{W_h C_h}{W_c C_c} = \pm \frac{t_{cL} - t_{cx}}{t_{hL} - t_{hx}} \quad (4-6)$$

This is true as the heat given by the hot substance equals the heat received by the cold substance, and the integration can be extended to the complete surface, from R to L , giving

$$3600 W_h C_h (t_{hL} - t_{hR}) = \pm 3600 W_c C_c (t_{cL} - t_{cR})$$

or

$$\frac{W_h C_h}{W_c C_c} = \pm \frac{t_{cL} - t_{cR}}{t_{hL} - t_{hR}} \quad (4-7)$$

From (4-5)

$$t_{cx} = \mp \frac{W_h C_h}{W_c C_c} (t_{hL} - t_{hx}) + t_{cL}$$

Hence,

$$(t_{hx} - t_{cx}) = \Delta t_x = \left(1 \mp \frac{W_h C_h}{W_c C_c} \right) t_{hx} \pm \frac{W_h C_h}{W_c C_c} t_{hL} - t_{cL} \quad (4-8)$$

$$d\Delta t_x = \left[1 \mp \frac{W_h C_h}{W_c C_c} \right] dt_{hx} \quad (4-9)$$

$$= \frac{t_{hL} - t_{hR} - t_{cL} + t_{cR}}{t_{hL} - t_{hR}} dt_{hx}$$

$$= \frac{\Delta t_L - \Delta t_R}{t_{hL} - t_{hR}} dt_{hx}$$

or

$$dt_{hx} = \frac{(t_{hL} - t_{hR})}{\Delta t_L - \Delta t_R} d\Delta t_x \quad (4-10)$$

where

$$\Delta t_L = t_{hL} - t_{cL} \quad \text{and} \quad \Delta t_R = t_{hR} - t_{cR}$$

This expression (4-10) is the same for parallel and for counter flow.

Now

$$U_x \Delta t_x dA = 3600 W_h C_h dt_{hx}$$

or

$$dA = 3600 \frac{W_h C_h dt_{hx}}{U_x \Delta t_x} = 3600 \frac{W_h C_h (t_{hL} - t_{hR})}{\Delta t_L - \Delta t_R} \frac{d\Delta t_x}{U_x \Delta t_x}$$

$$\int_0^A dA = A = \frac{q_t}{\Delta t_L - \Delta t_R} \int_{\Delta t_R}^{\Delta t_L} \frac{d\Delta t_x}{U_x \Delta t_x} \quad (4-11)$$

The unit transmittance U_x may be independent of Δt_x and constant, or it may vary with Δt as shown at (b) and at (c).

(a) If U_x is a constant, U , equation (4-11) becomes

$$A = \frac{q_t}{U(\Delta t_L - \Delta t_R)} \int_{\Delta t_R}^{\Delta t_L} \frac{d\Delta t_x}{\Delta t_x} = \frac{q_t}{U(\Delta t_L - \Delta t_R)} \log_e \frac{\Delta t_L}{\Delta t_R}$$

$$AU \frac{\Delta t_L - \Delta t_R}{\log_e \frac{\Delta t_L}{\Delta t_R}} = q_t$$

It is seen that, when $U_x = \text{constant}$

$$\frac{\Delta t_L - \Delta t_R}{\log_e \frac{\Delta t_L}{\Delta t_R}} = \text{mean } \Delta t \quad (4-12)$$

It is well to note that since (4-11) may be written as

$$A = \frac{q_t}{\Delta t_L - \Delta t_R} \int_{\Delta t_R}^{\Delta t_L} \frac{d(\log_e \Delta t_x)}{U_x} \quad (4-13)$$

the value of the integral may be found if values of U_x are known for values of Δt_x , although there is no known functional relation, by plotting a curve of $1/U_x$ against $\log_e \Delta t_x$ between Δt_R and Δt_L and finding the area between this curve and the axis of $\log_e \Delta t_x$.

(b) When $U_x = U' \div \Delta t_x^n$ another value of mean Δt results.

Substituting the value $\frac{U'}{\Delta t_x^n}$ for U_x of (4-11) there results on integration

$$A = \frac{q_t}{(\Delta t_L - \Delta t_R)U'} \frac{(\Delta t_L)^n - (\Delta t_R)^n}{n} \quad (4-14)$$

This gives

$$\frac{q_t}{A} = (U \text{ for mean } \Delta t)(\text{mean } \Delta t) = n \frac{(\Delta t_L - \Delta t_R)U'}{(\Delta t_L^n - \Delta t_R^n)} \quad (4-15)$$

Since $U \text{ for mean } \Delta t = \frac{U'}{(\text{mean } \Delta t)^n}$

$$\text{Mean } \Delta t = \left[\frac{n(\Delta t_L - \Delta t_R)}{(\Delta t_L)^n - (\Delta t_R)^n} \right]^{\frac{1}{1-n}} \quad (4-16)$$

(c) For the third case $U_x = U'(1 + b\Delta t_x)$ and (4-11) becomes

$$\frac{A}{q_t} = \frac{1}{U'(\Delta t_L - \Delta t_R)} \int_{\Delta t_R}^{\Delta t_L} \frac{d\Delta t_x}{\Delta t_x + b(\Delta t_x)^2} \quad (4-17)$$

This integral may be separated into the sum of two definite integrals (α) and (β).

$$(\alpha) = \int_{\Delta t_R}^{\Delta t_L} \frac{d\Delta t_x}{\Delta t_x} = \log_e \frac{\Delta t_L}{\Delta t_R}$$

and

$$(\beta) = \int_{\Delta t_L}^{\Delta t_R} \frac{b\Delta t_x}{1 + b\Delta t_x} = \log_e \frac{1 + b\Delta t_R}{1 + b\Delta t_L} = \log_e \frac{(1 + b\Delta t_R)U'}{(1 + b\Delta t_L)U'} = \log_e \frac{U_R}{U_L}$$

Moreover,

$$\begin{aligned} U'(\Delta t_L - \Delta t_R) &= U'\Delta t_L + bU'\Delta t_R\Delta t_L - bU'\Delta t_R\Delta t_L - U'\Delta t_R \\ &= \Delta t_L U'(1 + b\Delta t_R) - \Delta t_R U'(1 + b\Delta t_L) \\ &= \Delta t_L U_R - \Delta t_R U_L \end{aligned}$$

These give

$$\frac{A}{q_t} = \frac{1}{\Delta t_L U_R - \Delta t_R U_L} \log_e \frac{\Delta t_L U_R}{\Delta t_R U_L} \quad (4-18)$$

or

$$\frac{q_t}{A} = (U \text{ for mean } \Delta t)(\text{mean } \Delta t) = \frac{\Delta t_L U_R - \Delta t_R U_L}{\log_e \frac{\Delta t_L U_R}{\Delta t_R U_L}} \quad (4-19)$$

The value of mean Δt cannot be separated from this equation.

The order of Δt_L and Δt_R with U_L and U_R in any of the above equations may be changed in numerator and denominator, allowing the use of either end as R and the other as L . These equations may be used

when one of the temperatures remains constant, as with steam on one side, since the quantities needed are the Δt 's at the two ends. If both temperatures remain constant the expressions are indeterminate, but Δt remains the same over the whole area and is the value of mean Δt .

In the ordinary direct radiator, there is no appreciable change in temperature on the steam side and only a small change on the air side; in hot-water radiators there is a limited change on the water side, and in Vento heaters and aerofins there is a very appreciable change in the air temperature.

With known expressions for the values of U for these latter devices, values of mean Δt would be found by (4-12) or (4-16), or the value of q_r/A would be found by (4-19).

In general, however, each form of radiator is tested to give design data of such form as to simplify the solution of problems.

With the general methods of finding U and mean Δt for any problem known, the methods used in practice will be examined, and it is seen that usually the full value of $U(t_h - t_c)$ or $(U \text{ for mean } \Delta t)(\text{mean } \Delta t)$ is found by experiment and reported as heat per square foot per hour (q_a) or pounds of condensation per square foot per hour (W_c). Then

$$q_r = q_a A = W_c(h_s - h_o)A \quad (4-20)$$

h_s, h_o = enthalpies per pound of steam at supply and condensate at return.

Direct Radiators. For many years the value of U in the formula, for steam or hot water radiators,

$$q_r = UA(t_h - t_c)$$

was used with the experimental value 1.8 Btu per hr per sq ft per deg difference between the steam temperature, t_s , or the arithmetic mean of the temperature of the entering and leaving water, $\frac{t_e + t_o}{2}$, and the temperature of the room, t_r . In 1920, however, Allen and Rowley* showed, from the research at the laboratory of the A.S.H.V.E., that the coefficient U really included heat transmission by radiation as well as the heat conveyed from the radiator by convection in the air currents. This, of course, shows quite clearly that the distribution of radiator surface plays an important part in the average value of U for a given radiator, and in finding the value of U one must know this distribution of surface.

* Determination of Radiant Heat Given off by a Direct Radiator, J. R. Allen and F. B. Rowley, Trans. A.S.H.V.E., 1920.

The equation for transmission by radiation is

$$q_r = 0.172AE \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \frac{\omega}{2\pi} \right] \quad (3-1)$$

The term A refers to the area which can radiate heat. In any given radiator adjacent sections prevent radiation from the full surface, and so a study must be made to ascertain what fraction of the radiator surface can emit radiant energy. The angle ω for the complete radiator is of course 2π in this case, as the radiator is surrounded by the room.

In applying the results of these experiments, T_s is taken as the absolute temperature of the steam or water on account of the small temperature drop to the outer surface of the radiator, and T_r is used as the temperature of the room. By placing the radiator in a vacuum chamber, the walls of which were held at a temperature below that of the steam, this radiation transfer per square foot of total radiation surface was found to be

$$q_r = 0.157R \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \right] \quad (4-21)$$

in which R is the ratio of the enveloping area of the radiator to the total rated surface.

There yet remained the heat carried away by convection in the currents of air passing over the radiation surface, which was determined by the difference between the total heat delivered and the radiated heat. As the temperature of the air bathing the radiator increased as it passed upward over the surface, its ability to receive heat through the air film decreased with height, and the experiments showed that the quantity given to the air became less as the height of the radiator increased. The final formula for the average value of U resulting from this work is

$$U = \frac{0.157R}{(t_s - t_r)} \cdot \left[\left(\frac{T_s}{100} \right)^4 - \left(\frac{T_r}{100} \right)^4 \right] + C_c \quad (4-22)$$

C_c was given a value of 1 by Allen, although his experiments show that this value changes with height.

- $C_c = 1.100$ for radiators 20 in. and under in height
- $= 1.09$ for 26 in. height
- $= 1.025$ for 38 in. height
- $= 1.005$ for 45 in. height

$$R = 2 \frac{(\text{Width} \times \text{Height} + \text{Length} \times \text{Height} + \text{Length} \times \text{Width})}{\text{Rated surface in square feet}}$$

The value of 0.157 varied with the brightness of the radiator surface.

It is seen that the radiator of a single tube, or column of few sections, will be more effective per square foot than a wide radiator of the same number of sections, because R will be changed with the change of width or length. The height will affect U on account of the change in convection effect, although the value of R does not change appreciably with height.

Because the temperatures have an effect on the radiation term and therefore on the average heat flow, as seen by the equations above, a standard of these for testing direct steam radiators has been established by the A.S.H.V.E. as 215°F for the steam and 70°F for the room temperature. For convectors the room temperature is made 65°F while for hot-water radiators the mean $t_r = 170^{\circ}$ and $t_r = 70^{\circ}$ for direct radiators and 65° for convectors. The value of 170° for t_r when water is used is the average between the temperature of inlet and that of outlet.

The following values for U (including both effects) were found by the experimenters for radiators of 10 sections in length where the height and thickness were varied.

TABLE 4-3
VALUES OF U FOR DIRECT RADIATORS

Width of Radiator	Height			
	20 in.	26 in.	38 in.	45 in.
1 column.....	1 97	1 88	1.78
2 columns.....	1.82	1 73	1.64	1 61
3 columns.....	1 74	1 66	1.56	1.50
4 columns.....	1 63	1 55	1 45	1 41

Wall radiator combinations—2 14

Experiments to determine the change in average heat transfer with change in number of sections have shown that the average value of U for 20 sections is only 3% less than that for 10 sections, and when the number is reduced to 5 sections the mean value is increased by 6%. For a very short radiator of 2 sections, the average value of U is increased 27% over that for 10 sections.

In these experiments the value of U was computed by considering Δt to be $t_s - t_r$, since the temperature on each side was considered to be

constant. The value of $U(t_s - t_r)$ for a standard radiator is computed by using the coefficient 1.65 as an average and Δt , for the standard conditions for steam radiators, 215-70 or 145. This gives the value of q_s as 1.65×145 or 239.25 Btu per hr per sq ft. From this it is seen that the rating of a radiator in equivalent square feet can be obtained by dividing the heat liberated per hour under standard conditions by 240 Btu. This is common practice today, and radiator sections are so rated in equivalent square feet of direct radiation (EDR), each of which delivers 240 Btu per hr with steam at 215° F and the room at 70° F.

If radiators are rated at this standard transmission for 10 section radiators, regardless of length, the radiators will be 3% under size if they are 20 sections in length and 6% over size if only 5 sections in length. The tests of the radiators of different heights or depths to determine EDR will include any effect of height or number of columns.

Allen determined the effect of varying the temperature difference on the two sides of a 38 in. two-column radiator, $4\frac{1}{2}$ in. wide, and found that the value of U varied from 1.584 for $\Delta t = 100^\circ$ F to 1.616 for 120° F, 1.653 for 140° F. Later experiments have guided the A.S.H.V.E. in deciding that the coefficient of heat transfer for radiators varies as the 1.3 power of the temperature differences. Hence, to obtain actual deliveries for different conditions of room and fluid the following equations may be used:

$$q_{act} = 240 \left(\frac{t_s - t_r}{145} \right)^{1.3} \text{ for steam radiators} \quad (4-23)$$

$$q_{act} = 240 \left(\frac{t_s - t_r}{150} \right)^{1.3} \text{ for steam convectors} \quad (4-24)$$

$$q_{act} = 150 \left(\frac{\frac{(t_{ws} + t_{wa})}{2} - t_r}{100} \right)^{1.3} \text{ for water radiator} \quad (4-25)$$

$$q_{act} = \text{Actual Btu per hr per sq ft}$$

The values of the multipliers of q_{st} , or 240 Btu per sq ft per hr, to obtain the actual delivery at different room temperatures and different heating fluid temperatures are given in Table A-9. This appeared in the A.S.H.V.E. Guide for 1933.

Paint Effect. A number of experiments have been made to determine the effect of paint on the heat transfer from radiators. If one paint is placed on top of another paint, or several layers of paint, the effect pro-

duced is always that of the final coat, its nature and color. There is practically no difference in the value of U between that for bare iron as it comes from the foundry and that after dipping in a gray paint or giving one coat of dull black paint. Aluminum bronze or other lustrous paint reduces the value of U by about 10%, by reducing the part of the transfer due to radiation.

To apply the above formulae, the amount and size of a radiator to supply 7500 Btu per hr with steam at 5# gage to a room held at 75° will be determined. The space available for the radiator is 8 in. deep and 36 in. wide. The radiator is to be finished with aluminum bronze.

From Table A-9, page 404 the temperature conditions give a multiplier of 1.063 so that each square foot of radiation will deliver

$$240 \times 0.90 \times 1.063 = 229 \text{ Btu per sq ft per hr}$$

$$\text{Sq ft} = \frac{7500}{229} = 32.7 \text{ sq ft}$$

Using the chart, Table A-6, for either a 5 tube (8 in. wide) or 4 tube ($6\frac{5}{8}$ in. wide), the number of sections will be

7 sections, 5 tubes, 38 in. high, $17\frac{1}{2}$ in. long (35 sq ft)

9 sections, 5 tubes, 26 in. high or 4 tube 32 in. high, $22\frac{1}{2}$ in long ($31\frac{1}{2}$ sq ft)

12 sections, 4 tubes, 26 in. high, 30 in. long (33 sq ft)

12 sections, 5 tubes, 20 in. high, 30 in. long (32 sq ft)

The height and width available will determine which of this set may be used. If a length 36 in. were available for the radiator and valves, and low narrow radiation was advisable, the 12 section, 4 tube, 26 in. radiator would be used; if floor space were at a premium, the 7 section, 5 tube, 38 in. radiator would be selected.

Wall Radiators. By test these give 300 Btu per hr per sq ft of nominal surface when the bars are vertical; for radiation with horizontal bars, this is reduced 10%.

The room above would require 26 sq ft of wall radiation which would call for four No. 7A Peerless wall radiators (see page 93).

Enclosed Radiators. When standard radiators are placed behind grilles or enclosed by a cover the heat delivered is changed by the reduction of radiation and the variation in the convection effect by a different velocity of air flow. The A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings gives the following allowances, based on the experiments of Dr. C. W. Brabbée, for the arrangements of Fig. 83:

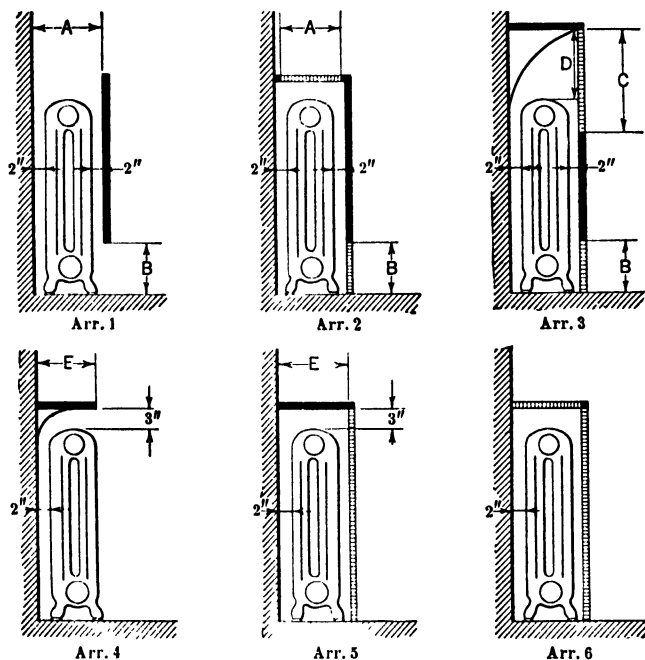


FIG. 83.—Types of Radiator Enclosures from A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilation of Buildings.

TABLE 4-4
ALLOWANCES FOR ENCLOSED RADIATORS

Type	Installation Conditions	Heat Emission Shall be Altered by Percentage Indicated
Arr. 1	When dimension <i>A</i> is as shown in Arr. 1 and dimension <i>B</i> is equal to 80% of <i>A</i>	10% increase
Arr. 2	When dimension <i>A</i> is as shown and dimension <i>B</i> is equal to 80% of <i>A</i>	5% increase
Arr. 3	When dimension <i>B</i> is equal to 80% of <i>A</i> (as in Arr. 1), dimension <i>C</i> is equal to 150% of <i>A</i> and dimension <i>D</i> is equal to <i>A</i>	No change
Arr. 4	When dimension <i>E</i> is equal to 50% of <i>A</i>	10% reduction
	When dimension <i>E</i> is equal to <i>A</i>	20% reduction
	When dimension <i>E</i> is equal to 150% of <i>A</i>	35% reduction
Arr. 5	When dimension <i>E</i> is equal to <i>A</i>	30% reduction
Arr. 6	When as shown	5% reduction

Enclosed Radiator. If arrangement 2, Fig. 83, is used for the radiator in the problem above, a 5% reduction should be made from the requirements of problem using black radiators. This would require 28 sq ft and 10 sections of a 4 tube radiator 26 in. high to be installed.

Fantom Radiators. The fantom flue radiators are listed in square feet per section which will deliver 240 Btu per hr per sq ft under standard conditions, and the selection of these is the same as that of direct radiators.

Pipe Coils. For pipe coils the radiation area is almost the full outside surface of the pipe, but when the pipes are arranged vertically above each other the upper pipes are not so effective for convection. With pipes in a horizontal plane, however, the distribution is most advantageous for both radiation and convection effects. Heat from pipe coils is computed by the formula:

$$q_t = UA(t_s - t_r) \quad (4-26)$$

the value of U being 2.6 for a single pipe or any number of pipes in a horizontal coil, while for coils in a vertical plane the value of U is 2.50 for 2 pipes, 2.15 for 4, 1.6 for 8 and 1.3 for 12 pipes.

For the room in the problem above, the length of a 8 pipe vertical coil of $1\frac{1}{4}$ in. pipe would be

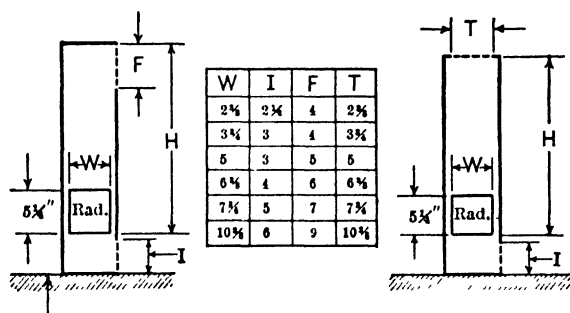
$$\frac{7500}{1.6 \times (227 - 75) \times 8 \times 0.43} = 8.95 \text{ ft length}$$

Concealed Radiators. When radiators are concealed in partitions or placed in separate cabinets and air is caused to flow over the radiator by pressure difference due to a fan, or to the chimney effect of a hot column of air above the radiator, the heat transfer is reported not as U but as a number of pounds of steam condensed per hour per square foot, the Btu per hour per square foot or the number of equivalent square feet of surface for a given complete radiator as used. The temperature rise in the outlet air as it leaves the radiator is also reported in many of these data sheets. With the rating and the temperature rise the engineer can select the radiator needed for a given heat demand and temperature of outlet.

Convectors. When special radiators of extended surface such as the Arco convectors are used the principal effect is that due to heat carried as convection by the air current, and the manufacturers give tables of equivalent square feet of surface based on delivering 240 Btu per hr per sq ft with steam, and 150 Btu per hr per sq ft for water when the air is

taken from a room at one temperature and returned to it at higher temperature. Such a table is given in Table A-8 for these Arco convectors. These ratings are obtained with an inlet opening at the bottom of 4-in. height and an upper grille as shown in Fig. 56. For one other type of extended radiator, the Murray, the ratings vary with the distance between the bottom of the convector and the outlet from the duct above it, as shown by the proportions given in Fig. 84, which must be followed to obtain the output given by the manufacturer.

Table A-8, for the Arco convector, contains values up to lengths of 48 in., although the full table of the manufacturer extends to 100 in. The rating is given in square feet, each of which delivers 240 Btu per sq ft per hr with steam at 215° F and room at 70° F, and 150 Btu per



Courtesy of American Radiator Co.

FIG. 84.—Dimensions of Enclosures for Murray Concealed Radiator.

sq ft per hr for water at 170° mean temperature. The transmission for other mean temperatures of water will be 1.7 Btu per degree difference from that at 170°.

In this table the stack height is measured from the bottom of the convectors, 3 in. above the 4-in. opening in the face, to the top of the enclosing duct channel or radiator cover from the 5-in. grille in the vertical face of which the air is delivered. The grille contains 60% free opening. If the outlet grilles are at the tops of the stacks there is an increased heat delivery of 15% for 11 in. stack heights and 6% for 15-in. stacks, and for 19 in. there is a 2% increase. For greater stack heights the horizontal outlet is no better than one in the vertical face.

The table given is for #5 units which contain three sets of finned tubes; to get approximate values of #3, two sets, #7, four sets, and #9, 5 sets, the proportional multiples 2/3, 4/3 and 5/3 may be used, although for accurate work the complete data of the American Radiator Company for the Arco convector must be employed.

As an illustration: The room requiring 7500 Btu per hr from one radiator placed in a flue which has a stack height, H , of 50 in. from the bottom of the unit would require

$$\frac{7500}{240} = 31.2 \text{ sq ft for steam at } 215^\circ$$

or

$$\frac{7500}{150} = 50 \text{ sq ft for water at } 170^\circ$$

This would require a #5 Arco convector $30\frac{1}{2}$ in. long with steam, or $45\frac{1}{2}$ in. long with water.

Nesbitt Radiators. Nesbitt radiators have been tested and the values of U at different frontal velocities of standard air have been determined for the logarithmic mean Δt . In this method the air actually used per minute is reduced to its volume at 70°F and 29.92 in., and then divided by the overall area of the face of the radiator to determine the frontal velocity, V , in feet per minute of 70° air. The area of the heating surface per square foot of frontal area is found and called A . Then

$$\frac{0.24}{13.35} \times 1 \times V \times 60(t_o - t_i) = UA \text{ mean } \Delta t = UA \frac{\Delta t_i - \Delta t_o}{\log_e \frac{\Delta t_i}{\Delta t_o}} \quad (4-27)$$

$$\Delta t_i = t_s - t_i$$

$$\Delta t_o = t_s - t_o$$

$$\Delta t_i - \Delta t_o = t_o - t_i$$

$$\frac{UA}{1.08V} = \log_e \frac{\Delta t_i}{\Delta t_o} = 2.3 \log_{10} \frac{\Delta t_i}{\Delta t_o} \quad (4-28)$$

$$\frac{\Delta t_i}{\Delta t_o} = \log^{-1}_{10} \frac{UA}{2.5V}$$

Hence

$$t_o - t_i = (t_s - t_i) \left(1 - \frac{1}{\log^{-1}_{10} \frac{UA}{2.5V}} \right) \quad (4-29)$$

The value of U is determined by (4-28) from experiment by varying the quantities of the equation and U is plotted against velocity V . Eq. (4-29) will then give the rise in temperature for a given A and V with values of t_s and t_i , and the value of mean Δt and the area for a given q , may be computed. The values of U will vary along a curved line, being 5.6 at 300 fpm, 7.45 at 600 fpm, 8.85 at 900 fpm and 9.25 at 1000 fpm.

The Nesbitt heaters are made with different arrangements of plates and tubes so that the value of A per square foot of frontal area varies as shown in table below:

TABLE 4-5
NESBITT RADIATORS

Series	Tubes	Plates	Fin Spacing, in.	Surface per Sq Ft of Frontal Area
72	7	3×8	$\frac{1}{8}$	51.9
73	7	3×8	$\frac{3}{16}$	35.3
74	7	3×8	$\frac{1}{4}$	27.0
42	4	$1\frac{1}{8} \times 8$	$\frac{1}{8}$	24.6
43	4	$1\frac{1}{8} \times 8$	$\frac{3}{16}$	16.8
44	4	$1\frac{1}{8} \times 8$	$\frac{1}{4}$	12.9

With steam at 215° and entering air at 10° F for the problem of this chapter the temperature rise in a #72 Nesbitt unit heater with 600 fpm will first be found, then the amount of air and surface. U for 600 fpm is 7.45.

$$\log^{-1}_{10} \frac{7.45 \times 51.9}{2.5 \times 600} = 1.81$$

$$t_o - t_i = (215 - 10) \left(1 - \frac{1}{1.81} \right) = 92^\circ \text{ F}$$

This means that the air is heated to 102° F and the amount of 70° air required to be circulated per minute to supply 7500 Btu per hr by cooling to 70° will be

$$\frac{7500}{60 \times 0.018(102 - 70)} = 217 \text{ cfm}$$

The mean temperature will be

$$\frac{205 - 113}{2.3 \log \frac{205}{113}} = 155^\circ \text{ F}$$

The total heat per hour to be transmitted by the radiator is

$$q_t = 60 \times 217 \times 0.018(102 - 10) = 21,600 \text{ Btu per hr}$$

The surface of the radiator is given by

$$A = \frac{21600}{7.45 \times 155} = 18.7 \text{ sq ft}$$

The area A per square foot of frontal area is 51.9 so that the frontal area will be

$$A_f = \frac{18.7}{51.9} = 0.361 \text{ sq ft} = 52 \text{ sq in.}$$

From air requirements

$$A_f = \frac{217}{600} = 0.362 \text{ sq ft}$$

This checks the above.

The length of this A #72 Nesbitt radiator would be

$$\frac{52}{8} = 6.5 \text{ in.}$$

Of course such a short Nesbitt radiator is not manufactured. The method of computation has been explained, however, as an application of theory.

Forced Blast Air Heaters. Experiments have been made on Vento heaters, pipe coils and aerofin coils, to determine the heat delivery and resulting temperature of air from one or more units in series, with different velocities of air flow and different temperature conditions of steam and entering air. The purpose of these experiments was to determine the effect of velocity and temperature difference on the performance of such heaters. The amount of heat given to the air will increase as the velocity increases, owing to the thinner air films and greater turbulence, but in this case the mass of air passing the heating surface increases at a faster rate than that of the heat delivery, so that the temperature of the delivered air decreases. It is also found that, when the air is passed over a number of similar units in a series arrangement, the amount of heat delivered by each successive unit of the same size, as well as the increase of air temperature, becomes less and less because Δt from steam to air for each successive unit is smaller, owing to the increasing temperature of the air. For this reason the results of tests are given on data sheets or by curves for definite numbers of sections in series showing the temperature after passing over one, two, three or more units, and the average heat delivered per rated square foot of this series or per standard unit of the series as the velocity or the entering air temperature is changed. Of course in this work the actual surface of the sections might be used for area, so as to compare units of different types, but it makes no difference in the application of the tables if nominal square feet or any other units are used in reporting the performance of some unit of area if in applying the results from these data sheets the amount of surface is computed in terms of these same units of area.

In making tests on these surfaces the amount of heat delivered is found by weighing the condensation and determining the enthalpy per pound, h_s , for the entering steam at its absolute pressure and quality and the enthalpy per pound of the condensate, h_c , leaving the heater. If W is the actual condensate in pounds per hour, or W_e the equivalent

condensation of some standard condition in pounds per hour for the standard enthalpies h_{ss} and h_{cs} there results:

$$q_t = W(h_s - h_c) = W_s(h_{ss} - h_{cs})$$

From this expression the actual W may be reduced to standard W_s and by dividing q_t or W_s by the heating surface of the heater, in actual or nominal square feet, the Btu per hour per square foot or the condensation per hour per square foot of actual or nominal area is known under the conditions of velocity and entering air temperature. The manufacturer always states in his data sheet the number of square feet of heating surface which has been used as the unit for the tests.

The velocity of the air changes as the air is heated so that the velocity reported for the test must be that occurring when the air has a certain temperature and is passing through a certain part of the apparatus. It was mentioned on page 111 that the coefficient of heat transfer depends on the mass velocity, wV or G , so that this quantity will remain constant for a given area as the air is heated, the increase of the velocity, V , being offset by the decrease in the density, w . Again, the area for air flow changes in passing through different parts of the apparatus. The area used in computations and reported in the data sheets can be that at any point of the unit, so long as the same place is used for the unit which is being selected in any design problem. The distribution of surface as well as that of the various stacks or frames in the application is always similar to that of the tests when these data are used in design. For instance, in the application of Vento data, the area used is the net area between sections of the stacks as tabulated in the Appendix and not the area of the casing occupied by the air entering the Vento heater or that enclosing the stack, while for the Aerofin heaters the clear frontal area of the inside of the frame around the tubes (the area of the flue leading to the heater) is used for area. With the area to be used in determining velocity fixed, the remaining factor is the volume per minute, and this is figured as the volume of the weight of air used per minute when this air is at 70° F. The air may never be at 70° F, but its volume at 70° F is computed and the velocity in feet per minute is obtained by

$$\text{Velocity in fpm} = \frac{\text{Vol per min at 70° in cu ft}}{\text{Test area of air channel in sq ft}} \quad (4-30)$$

This formula is used in practice to determine the area of air channel since the amount of air is known and the velocity desired in the installation has been selected.

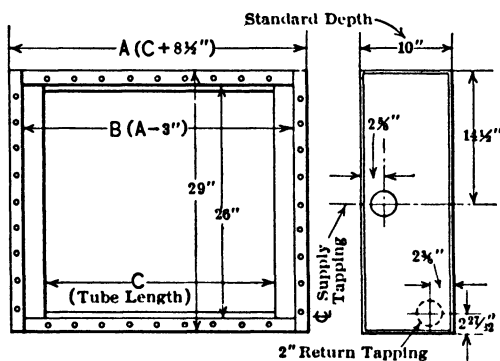
$$\text{Area of air channel in sq ft} = \frac{\text{Vol of air per min at } 70^{\circ} \text{ in cu ft}}{\text{Velocity selected in fpm}} \quad (4-31)$$

The area resulting from this is then used in connection with the data sheets of the particular manufacturer to fix the number of units required for this frontal area of the heater.

With this explanation, the following limited data are given from a number of manufacturers, reference being made to their complete tables for other information.

Aerofin Heaters. The Aerofin Corporation presents data from which Table A-11 is taken for one type of their product, flexitube Aerofin heaters. This gives average condensation per square foot and final temperature of air at different frontal velocities, when entering at different temperatures.

The flexitube Aerofin units are made with a frame of the standard thickness of 10 in. which may contain one row of eighteen $\frac{5}{8}$ -in. Aerofin



Courtesy of Aerofin Corporation

FIG. 85.—Dimensions of Casing of Flexitube Aerofin Units.

6 ft to 10 ft by 1 ft intervals. The over-all width or height of these units is 29 in., and the over-all length is $8\frac{1}{2}$ in. greater than the length of the tubes. The thickness or depth of the casing, as stated before, is 10 in. for all units. Flanges are formed at the edges for the attachment of casings. These dimensions are indicated in Fig. 85.

There are five different arrangements of tubes designated by numbers: #61 refers to tubes with 6 turns per in. of $\frac{1}{4}$ in. fins and one row of tubes in depth. Units of seven $\frac{3}{8}$ fins per in. and one row of tubes in depth are called #71 units; two such tubes in depth are called #72; #81 has eight $\frac{1}{2}$ fins per in. with one row of tubes, and #82 has two.

tubes, or two such rows with tubes staggered. The fins are $\frac{1}{4}$, $\frac{3}{8}$ or $\frac{1}{2}$ in. wide and are put on with 6, 7 or 8 turns of the helix per inch. The variations are made to give different outputs to the units. The space between the inside faces of the casing or width is about $25\frac{1}{8}$ in. for all units. These units are made of varying lengths of tubes, 2 ft to 6 ft by 6 in. intervals, and from

The velocity of air flow which is used in reporting test results, and which must be used in applying data, is the result of dividing the volume of the air per minute in cubic feet as if it were at 70° F and of standard pressure by the frontal area of the units for air flow, and not by the area between tubes. This area is equal to the height $25\frac{1}{8}$ in. for a single row and 26 in. for a double row multiplied by the length of tubes less $\frac{1}{2}$ in. The areas for air flow and the nominal square feet of heating surface for each length of units are given in various tables of the data book, and those for #61, #71 and #72 units are given in Table A-10.

The velocities through these heaters, being for the total frontal area, are lower than values given for heaters in which velocity is figured on area between sections as in the Vento data, or between pipes as in pipe coils. In these cases the net area is about one-half the frontal area used with Aerofin units. The manufacturers of Aerofin units recommend velocities of 600 to 800 fpm with two rows of pipes and 400 to 500 fpm when the heater has eight rows of tubes in depth. These velocities are for public building work; for factory work the velocity will be increased by 200 fpm. The drop in pressure due to friction of air through the heaters is given in Table A-12.

The tappings for steam inlets and drips are made of nipples attached to the steam headers at the ends. There is one steam nipple of 2 in. pipe for #61 units up to 6 ft length, for #71 and #81 units up to $4\frac{1}{2}$ ft, for #72 units up to $3\frac{1}{2}$ ft and for #82 units up to $2\frac{1}{2}$ ft. Above these lengths the nipple is of 3 in. pipe. The drip nipples are of 2 in. pipe.

The performance of these various units is stated in the data book for various steam pressures, and the table for #61, #71 and #72 Flexitube Aerofin with steam at #5 gage has been selected for reproduction in Table A-11.

In applying such tables a desired velocity is taken and then the number of rows of tubes is found to give, at this velocity, the desired final temperature with air entering at a given temperature. The average condensation per square foot of heating surface per hour with steam and condensate at 227° F is also included. By reducing the air needed to volume at 70° F and dividing the volume in cubic feet per minute by the selected velocity, the total frontal area is determined and the table of dimensions will give size and number of units and their length or the sum of the length of units to give the necessary area if more than one unit is required. The number of rows of tubes for the temperature of outlet fixes the number of units which must be used in series.

Some engineers stop at this point. The author recommends, however, that a check be made for the equality of the heat required for the

air and the heat given up by the total heating surface from the condensation of steam at 227° F, as given in the table.

As an example, suppose that 2000 cfm of air at 30° with 50% relative humidity is to be heated with 5% steam to 80° F for an office building.

On referring to the table it is seen that the one section of #71 will heat 30° air to 79.2° F with a velocity at 70° of 400 fpm, and if 80° is actually desired, the velocity, as determined by interpolation, will have to be reduced to 386 fpm. This velocity is very low but must be adopted if the full surface of the heater is used. By interpolation, or by the construction of curves, the condensation will be 1.59 lb of steam per sq ft per hr at this speed.

Reference is now made to Chapter II for the heat required for the air and its moisture as well as its volume

$$W = \frac{2000 \times 60}{12 \times 34 + 1/2(0.07)} = 9700 \text{ lb per hr}$$

From the chart the moisture per pound of dry air is 12.01 grains and the wet-bulb temperature is 25° F.

Relative humidity at 80° F from the chart is under 10%, and from moisture content it is approximately $12.0 \div 155.8$ or 8%, and the wet-bulb temperature would be 51.6° F.

Enthalpy per pound of dry air entering from wet-bulb temperature of 25° F = 8.94 Btu.

Enthalpy per pound of dry air leaving at 51.6° F wet-bulb temperature = 21.23 Btu.

$$q_t = 9700(21.23 - 8.94) = 119,000 \text{ Btu per hr}$$

$$= 124.0 \text{ lb of condensate at } 227^\circ \text{ F.}$$

This then requires $124.0 \div 1.59$ or 78.0 sq ft of heating surface.

The volume of this air at 70° with 12 grains of moisture per pound of dry air will be $13.35 + 0.11 \times 0.04$ or 13.36.

The frontal area will then be

$$\text{Net area for air} = \frac{9700}{60} \times \left(\frac{13.36}{386} \right) = 5.63 \text{ sq ft.}$$

It is seen that #71 Flexitube with 2½ ft tubes is a little under size for heating surface and frontal area (70 sq ft of heating surface and 5.14 sq ft of frontal area). A 3 ft length gives 84 sq ft heating surface and 6.18 sq ft of net area, which is slightly large. Of course the larger size must be used. The resistance to air flow will be 0.05 in. of water.

The Aerofin Corporation builds special water coils for surface cooling and special tube units for surface cooling by the evaporation of volatile fluids to be used in cooling and dehumidifying air.

Their data sheets for the latter type give the tons of refrigeration per square foot of face area and the wet-bulb temperature of the leaving air for one or more tubes for series air flow for various velocities of equivalent air at 70° F with 50% relative humidity and for different entering

wet-bulb temperatures. Data are furnished for different temperatures of evaporation of the refrigerants used. Curves are shown from which the leaving dry-bulb temperatures may be found from entering dry- and wet-bulb temperatures for different velocities and numbers of rows.

To determine the size of refrigerant valve for each row of tubes, a table gives the percentage of the total refrigeration obtained from each successive coil.

The amount of equivalent standard (70° F, 50% relative humidity) air to be treated being known, a face velocity is assumed and the number of square feet of face area may be found; after selecting the unit nearest to this size, the actual velocity of standard air is known. With this and the entering wet-bulb temperature, the number of rows in series and the average refrigeration per square foot of frontal area can be found to give a desired leaving wet-bulb temperature.

This tonnage should then be checked by the amount of heat (sensible and latent) which should be removed from the air, remembering that 1 ton of refrigeration means a removal of 200 Btu per min.

The data book is to be consulted for full information.

In the continuous Aerofin water coils, eighteen tubes in parallel receive water which makes several passes to the outlet and is arranged to give a water flow counter to that of the air. Units are built in the standard 10-in. casing with different numbers of banks of tubes from two to six.

In computing the surfaces for air cooling, the volume of 50% R.H. standard air (70° F) is divided by the frontal or face velocity of not above 500 to 600 fpm, so that water may separate from the air stream. This gives the frontal area required, and the nearest unit for this area is found from Table A-27. With the exact frontal area, the velocity for the required air volume is determined.

The amount of water used in the cooler is fixed by the desired increase in water temperature and is found by dividing the heat removed from the air to bring its enthalpy from that of entrance to that of exit by this rise in water temperature. The gallons of water per minute (gpm) is found by dividing this heat removed by 500 times the temperature rise. From this quantity of water in gallons per minute the velocity through $\frac{5}{8}$ -in. tubes (0.05 in. walls) in parallel is found by dividing the quantity 1.235 gpm by the number of tubes in parallel (18 for each unit of frontal area). This velocity is required to determine the pipe friction for which water pressure is needed as well as to determine the heat transfer.

In practice it is found that these units deliver air at a point on the desired wet-bulb line of a psychrometric chart at which it cuts a straight line connecting the entering air condition and a point on the saturation

line at a temperature which is a mean between the desired wet-bulb and that of the entering water. (This is the mean temperature of air and water at the point of air discharge for counter-current flow.)

The difference of this dry-bulb temperature at exit just determined and that of the entering water gives Δt_2 for heat transfer, while the difference between the dry-bulb at entrance and that of the leaving water gives Δt_1 , for which

$$\text{Mean } \Delta t = \frac{\Delta t_1 - \Delta t_2}{\log_e \frac{\Delta t_1}{\Delta t_2}} \quad (4-32)$$

The values of heat U' per square foot of frontal or face area per row of tubes per degree of logarithmic mean Δt from the sensible heat q ,

$$0.241W_c(t_{de} - t_{di}) = q, \quad (4-33)$$

t_{de} = temperature dry-bulb entrance,
 t_{di} = temperature dry-bulb exit,

have been experimentally determined for different velocities of air and water and plotted in Table A-26. The correction due to the presence of water of condensation is also given. This correction varies with water velocity and with the ratio of the sensible heat removed from the dry air to that actually removed from the air.

$$\text{Ratio} = \frac{0.241(t_{de} - t_{di})}{h_e - h_i} \quad (4-34)$$

h_e = enthalpy associated with 1 lb of air at entrance;

h_i = enthalpy associated with 1 lb of air at exit.

To use these charts the ratio of the sensible heat removed to the actual heat removed is found. The velocity of standard air per square foot of frontal area and that of the water are used to determine the value of U' , Btu per hour per square foot of face area per row of tubes per degree total logarithmic temperature difference. This value of U' is then multiplied by the coefficient for sensible heat ratio and the value of mean Δt , and the product is divided into the sensible heat per square foot of frontal area to be removed to determine the number of rows in series to insure the result. Since the units are built with two to six banks of tubes in series the designer may make any desired combination.

A problem using this type of unit is given on page 293.

Vento Heaters. The dimensions of the Vento heater sections are given in Table A-13, as well as the square feet per section. The tappings are 2 in. for 30 in. sections, $2\frac{1}{2}$ in. for 40, 50 and 60 in. sections, and 3 in. for 72 in. sections. The manufacturer provides tables of ratings of surface and free area for each of these sections, similar to Table A-14, for the 60 in. section. This table gives data for stacks made up of different numbers of sections in one stack. Reference is made to the data book for other sizes.

The net air space in Table A-14 is the area between sections by which the velocity of 70° air has been computed. Inasmuch as this area is less than the frontal area of the Aerofin practice higher velocities will be found in the tables for temperatures and condensations.

A table of performance for the 5-in. centers with steam at 5% gage has been prepared from the data book with entering air temperatures at 0° and above. It appears as Table A-15. For other center distances and sizes, and for other entering air temperatures, as well as for steam of 30% gage, reference must be made to the complete data book.

The friction of air through Vento heaters in inches of water is given in Table A-16.

These tables are applied to a given problem:

Air used 10,000 cfm at 0° F dry
 Final temperature 160° F
 Discharge temperature 160° F
 Weight dry air 51,800 lb per hr
 Heat per hour = 2,000,000 Btu
 Volume of air at 70° = 11,520 cfm

Try 5 in. centers and 1200 fpm, Table A-15.

By plotting curves for temperature and condensation for extrapolation beyond the table limits, 9 stacks will be needed to give a temperature of 160° from zero air, with which there will be an average of 1.34 lb of steam condensed per hr.

$$\text{Total heating surface} = \frac{2\,000\,000}{960.6 \times 1.34} = 1555 \text{ sq ft}$$

$$\text{Heating surface per stack} = \frac{1555}{9} = 172 \text{ sq ft}$$

$$\text{Net area} = \frac{11\,520}{1200} = 9.6 \text{ sq ft}$$

For 60 in. sections the area of heating surface per stack requires slightly less than 11 sections, which have a net area of 10.13 sq ft. These are not close enough in area so that $4\frac{3}{8}$ in. centers should now be investigated, using the data book. This reference calls for 8 stacks to give the temperature, and the condensation is 1.31 lb per sq ft.

$$\text{Total heating surface} = 1555 \times \frac{1.34}{1.31} = 1590 \text{ sq ft}$$

$$\text{Heating surface per stack} = \frac{1590}{8} = 198 \text{ sq ft}$$

This requires a little over 12 sections, with a net area of 9.37 sq ft, which is sufficient. The design will then call for 8 stacks, each of 12 sections. This will provide 1536 sq ft of surface requiring $60\frac{1}{8}$ in. height, $57\frac{1}{2}$ in. wide and 82 in. in length. The weight would be 12,600 lb.

The friction would be 0.666 in. of water.

The method of checking the heating surface as well as net area must be used to select the proper center distance. It is also to be remembered that by plotting values of temperature or condensation the tables may be interpolated or extrapolated.

Blast Pipe Coil Heaters. The performance of the pipe coil heaters, which were extensively used in the past, is indicated by curves of final temperature and heat in Btu per lineal foot at various velocities of 70° air figured on clear space between pipes and for various sections four tubes in depth. The curves are shown in Figs. 86 and 87. The dimensions of a set of Buffalo Forge Company heaters are stated in Table A-17, which contains the areas for air passage used for velocity determinations. The friction loss in air flowing through the heater is also indicated in the table.

The problem above may now be solved for pipe coils.

Temperature range 0° F to 160° F; velocity 800 fpm. The curves for air entering at 0° F do not go to 160° F so that it will be necessary to combine curves. From the curves with entering air at 40° F it will be seen that 8 sections are required to heat the air to 160° F, and from curves with entering air at 0° F, $1\frac{1}{2}$ sections will be required to bring the air to 40° F. This means $9\frac{1}{2}$ sections. The average heat in Btu delivered per lineal foot per hour from these curves is respectively 380 for the 8 sections and 675 for $1\frac{1}{2}$ sections. (A half size section has two rows of tubes.)

The average heat from these will be

$$\frac{380 \times 8 + 1\frac{1}{2} \times 675}{8 + 1\frac{1}{2}} = 430 \text{ Btu per lin ft per hr}$$

$$\text{The total length of 1 in. pipe will be } \frac{2\,000\,000}{430} = 4650$$

$$\text{The length per section} = \frac{4650}{9.5} = 490 \text{ lin ft}$$

$$\text{The area for air flow} = \frac{11\,530}{800} = 14.4 \text{ sq ft}$$

It is seen that the section 3C is just under these quantities, having 477 lin ft and 14.0 sq ft of area. This would give a heater $4\frac{1}{2}$ ft wide \times 6 ft 10 in. high and 7 ft 1 in. long, weighing about 10,000 lb.

The friction loss in this heater would be $9\frac{1}{2} \times 0.061$ or 0.58 in. of water.

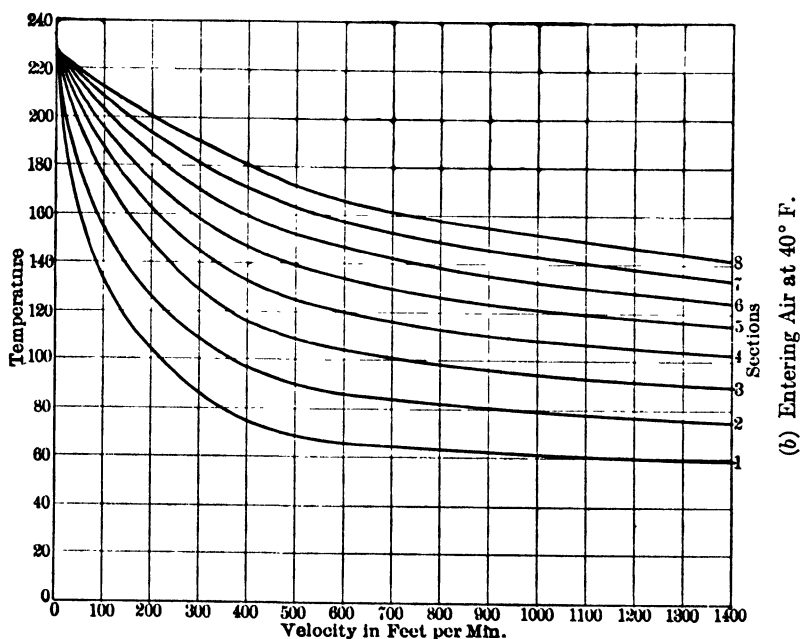
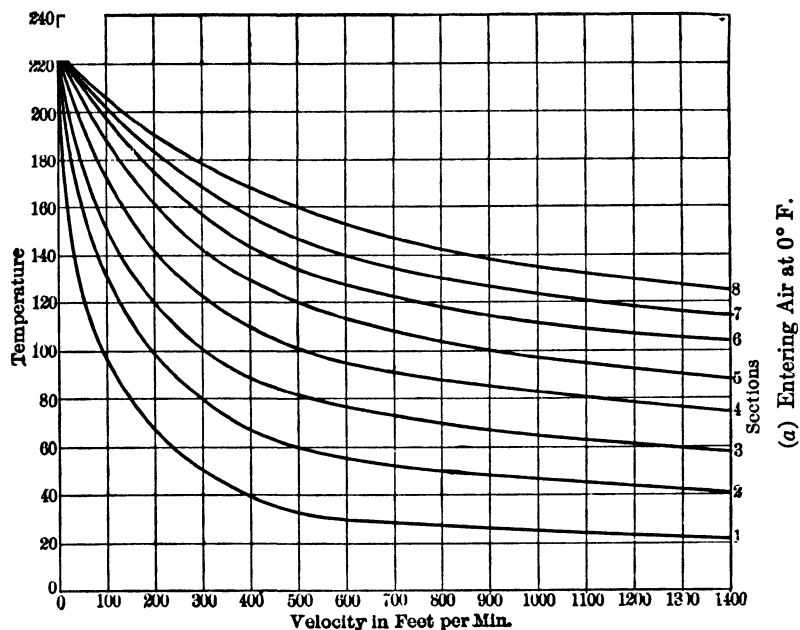
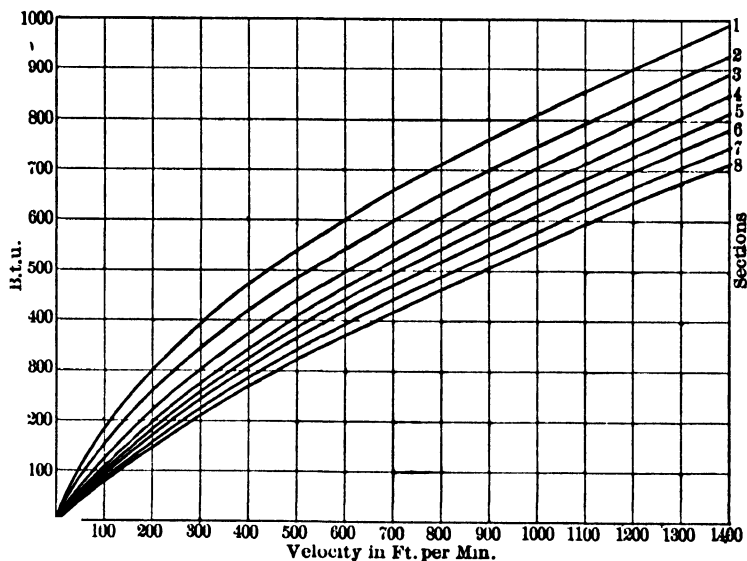
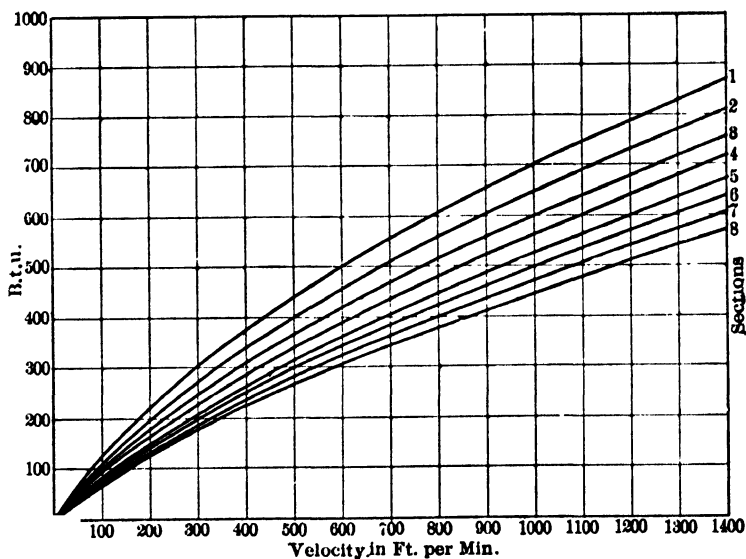


FIG. 86.—Outlet Temperature from Buffalo Heater Sections of Four Coils—Steam at 5 lb Gage and Velocity of Air Computed for 70° F Air through Tabular Air Passage Area. (Drawn from Data of Buffalo Forge Co.).



(a) Entering Air at 0° F.



(b) Entering Air at 40° F.

FIG. 87.—Heat Transmitted per Lineal Foot of One-Inch Pipe of Buffalo Heater Sections of Four Coils—Steam at 5 lb Gage and Velocity of Air Computed for 70° F. Air through Tabular Air Passage Area. (Drawn from Data of Buffalo Forge Co.).

Interpolating from Tables and Charts. The tables and charts can be used for inlet temperatures differing from those given by realizing that if the inlet air temperature t , higher than a tabular inlet temperature can be obtained from this tabular inlet temperature t_2 at the desired velocity by N_1 sections or units, and that the outlet temperature t_3 for this tabular temperature, t_2 , and the desired velocity, can be obtained from N_3 sections, then $N_3 - N_1$ sections will give the outlet temperature t_3 from the inlet temperature t , at this velocity.

The average condensation or heat transmission, C , is found by obtaining C_1 and C_3 at these tabular points and computing the desired mean C by

$$C = \frac{N_3 C_3 - N_1 C_1}{N_3 - N_1}$$

Pin Radiators. The radiators, Fig. 69, are used for natural draft indirect heating, as shown in Fig. 2. In such radiators the temperature resulting and the heat transfer depend on the velocity of air flow and temperature of the entering air. The air flow is usually controlled by the length and size of flues and fittings connecting the warm air outlet in the building to the outside. The flue can be designed for a given flow of air so that in selecting radiators the air flow in cubic feet per hour of 70° air per square foot of radiation can be used with the results of the tests by John R. Allen, as reported by Harding and Willard and shown in Fig. 88. Curves at 150 cfh per sq ft of radiation for use in residences and 200 cfh per sq ft of radiation for schools are given for both outlet temperature and heat per hour per square foot of radiation at various temperatures of entering air.

The curves for heat transmission are based on the formula

$$q_t = U \left(t_s - \frac{t_o + t_i}{2} \right)$$

If 200 cfh is used per sq ft with 1 in. pin radiators with entering air at 30° the final temperature will be 136.4° F and the heat will be 391 Btu per hr per sq ft.

If the heat required from the heater at this point of the building is 24,000 Btu per hr the surface required would be

$$\frac{24\,000}{391} = 61.4 \text{ sq ft}$$

This will require three of the Sanitary School Pin Radiator units.

Unit Heaters or Conditioners. The heat transfer for different quantities of air from the unit heater or air conditioners as well as the power demands, weight and space requirements are given in the data

books of the various companies and reference must be made to them after determining the needs of a given building space. Problems on the use of unit heaters will be found in a later chapter.

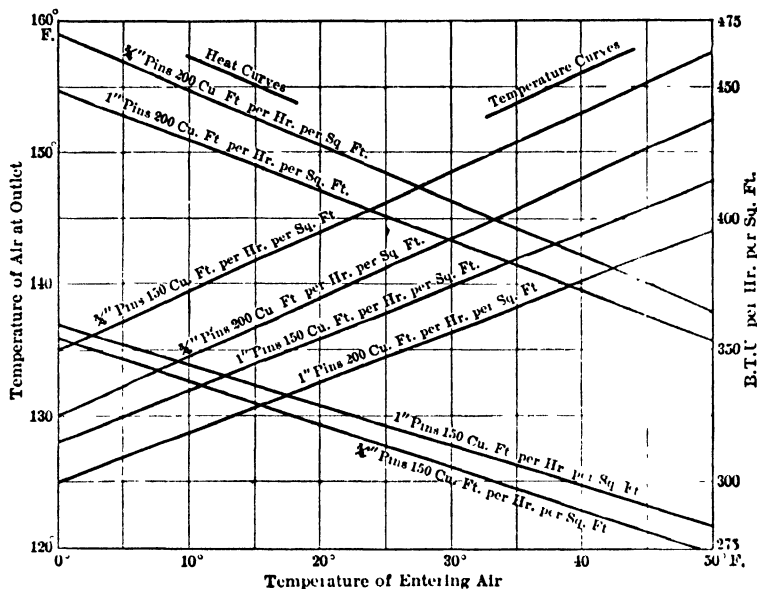


FIG. 88.—Performance of Pin Radiators.

PROBLEMS

1. A room held at 65° F with steam at 222° F has two windows 36 in. wide with sills 28 in. from the floor. The heat demand for the room is 10 Mbh. Select for this room a direct Corto radiator equipment, with bronze paint. Select the wall radiator equipment with brown paint. Select the size of Fantom radiators with white paint. Select the Arco equipment for the room, if units are placed in channels, with outlets 60 in. above floor. Sketch detail for architect giving sizes of grilles and flue dimensions. If the Corto radiators are used with an enclosure of arrangement 1, Fig. 83, what surface should be installed?

2. A factory building has a room for which $q = 600$ Mbh. It is desired to use $1\frac{1}{2}$ in. pipe radiators on walls with steam at 200° F and a room temperature of 50° F. Wall spaces for 40 such radiators are available, the maximum height being $4\frac{1}{2}$ ft and maximum length 15 ft. What length would you use for each leg of radiator, Fig. 59.

3. If the heat for a shop is to be delivered by 12,300 cfm of 70° air heated from 50 to 150, select the flexitube units to give this performance. Select the Vento units to give this performance. Select the pipe blast coil to give this performance.

4. A museum requires 280 Mbh added to an air supply at 30° F to Nesbitt unit heaters, Series 73. The steam temperature is 227° F, and it is desired to use a frontal velocity of 300 fpm. Find the total frontal area of radiators to be used.

5. Select the pin radiators to be used in problem 4.

CHAPTER V

METHOD OF CALCULATING HEAT REQUIRED FOR ROOMS

In practice, the method of finding heat losses depends on the importance of items which the engineer desires to employ in his computations. In any event the data should be recorded and the computations should be made for each room in a permanent manner and with such a methodi-

[illegible]

FIG. 89.—Spangler Data Sheet.

cal arrangement that results may be checked quickly and simply. The late Professor H. W. Spangler used a filing card form of data sheet, Fig. 89, which contained the complete data for one room. The author uses a larger 8×11 sheet, Fig. 90, on which practically the same data are recorded but the sheet is sufficient for several rooms.

HEATING DATA

DESIGN #620

RECORDER XYZ COMPUTER ABC DATE 11-16-35

Room No.	Sketch	Height Volume Occ.	Leakage	Areas		ΔT	U		Heat Demand		Occ. Heat Sun Effect	Collected Data		
				Total	Open- ings		Net Wall	Open	Wall	Open			Wall	Total
101		9' 0" 3110 Res.	W 60 ft 10.2 cfh per ft 612 cfh	N	198	28	170	70	1.13	0.29	2215	3450	5665	Heat demand..... 19 740 Occ. heat..... 19 740 Net heat..... 19 740 Special allowance... none Requirements..... 82 sq ft Install 2 12-sect. 5 tube 26"..... 84 sq ft Ratio..... 38.0-37.0
				E	63	28	35	70	1.13	0.29	2215	710	2925	
				S	141	65	76	70	1.13	0.29	5140	1545	6685	
				W	345	34	0.30	3520		
				F	345	5	0.10	175		
				L	612	70	0.018	770		
Special Notes: Glass large on west														
Ventilation requirements: none														
102		9' 0" 2160 Res.	W 58 ft 10.2 cfh per ft 592 cfh	N	0	Heat demand..... 14 275 Occ. heat..... 14 275 Net heat..... 14 275 Special allowance... none Requirements..... 59 sq ft Install 2 9-sect. 5 tube 26"..... 63 sq ft Ratio..... 36.6-34.3
				E	126	34	92	70	1.13	0.29	2680	1870	4560	
				S	134	56	98	70	1.13	0.29	4430	1980	6410	
				W	239	34	0.30	2440		
				F	239	5	0.10	120		
				L	592	70	0.018	745		
Special notes: Glass large on west.														
Ventilation requirements: none														
201		8' 6" 2910 Res.	N 24 ft 15.5 cfh per ft 15 ft 10.2 cfh per ft 525 cfh	N	178	35	143	70	1.13	0.20	2770	2000	4770	Heat demand..... 15 170 Occ. heat..... 15 170 Net heat..... 15 170 Special allowance... none Requirements..... 63 sq ft Install 2 11-sect. 5 tube 23"..... 66 sq ft Ratio..... 46.2-44.1
				E	68	28	40	70	1.13	0.20	2220	560	2780	
				S	138	22	116	70	1.13	0.20	1740	1620	3360	
				W	342	-5	..	0.19	-170	
				F	342	58	3770		
				C	342	70	0.018	660		
Special notes:														
Ventilation requirements: none														
205		8' 6" 1710 Res	N 24 ft 15.5 cfh per ft 372 cfh	N	140	22	118	70	1.13	0.20	1740	1650	3390	Heat demand..... 8740 Occ. heat..... 8740 Net heat..... 8740 Special allowance... none Requirements..... 36.4 sq ft Install 2 8-sect. 5 tube 23"..... 36 sq ft Ratio..... 47.0-47.5
				E	104	20	84	70	1.13	0.20	1580	1180	2760	
				S	0	
				W	201	-5	..	0.10	-100	
				F	201	58	..	0.19	2220	
				C	201	70	0.018	470		
Special notes:														
Ventilation requirements: none														

Fig. 90.—Data Sheet.

To apply the author's methods of calculating the complete heat requirement of a building, the plans of a residence with a small number of rooms will be chosen, so that all parts of the building may be considered. The same methods would be applied for any building, be it a school, office building, hotel, apartment house or factory.

The complete floor plans of the building from foundations to roof, the elevations with heights and window areas, sections with wall and

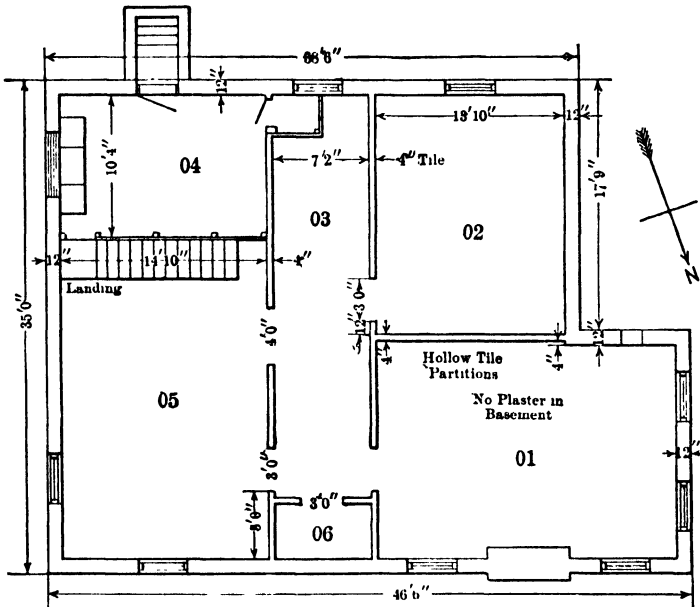


FIG. 91.—Cellar Plan.

floor construction and even detail drawings are needed in this study. The engineer should be given any information affecting heating or ventilation which the architect plans to place in the specifications rather than on the plans. These drawings may be the penciled drawings of the architect, as heating data to be placed on the final plans must be supplied to the architect by the heating engineer, but they must be sufficiently exact to give all data necessary for the calculations.

Such a set of plans is given in Figs. 91 to 95. The attic and roof plans are omitted as there were no attic partitions and all roof information is found on the drawings given. As plans are usually drawn with the front of the building facing the bottom of the sheet, the engineer must mark the directions of the points of the compass on plans to fix

the exposure of the various walls. From the nearest Weather Bureau station average monthly temperature, lowest and highest temperatures, average monthly humidity and direction and intensity of prevailing wind may be obtained for the locality in which the building is to be erected. Most of these stations publish yearly average weather data which will aid the designer. Reference may be made also to the tables in the 1936 A.S.H.V.E. Guide for summer and winter conditions and

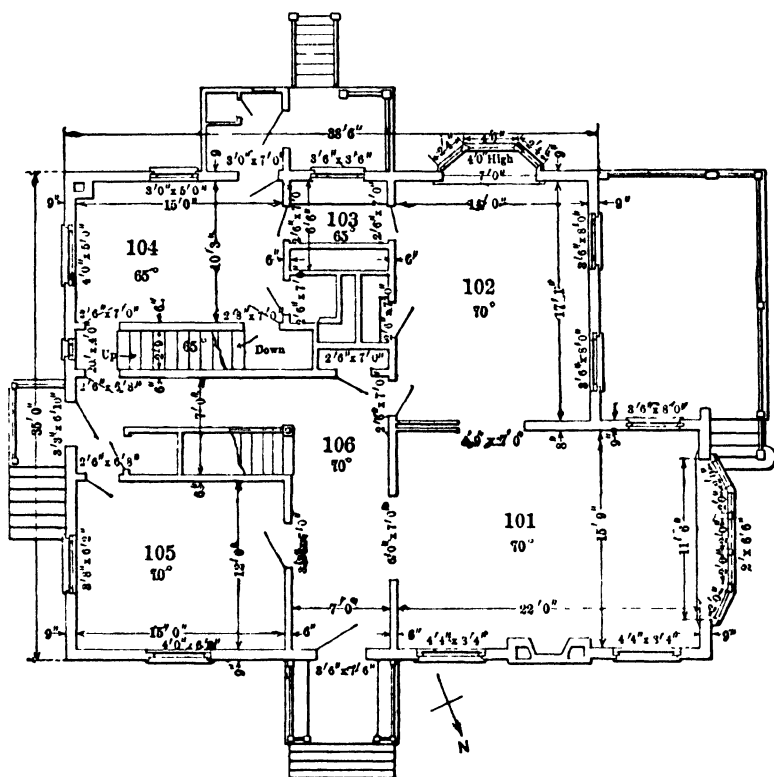


FIG. 92.—First Floor Plan.

to the magazine Heating and Ventilating for data on many localities in the United States and Canada. From these data the outside temperature and the direction and velocity of the wind for the heating season and air-conditioning season should be noted on the general data sheet, Fig. 96, as well as the orientation of the building. It is well to give each room a number, and if this has not been done by the architect the engineer should do it, giving to the first floor numbers between 101 and 199, to the second floor 201 to 299, to the twelfth

The differences of the temperatures on the two sides of each face are entered in the column marked Δt .

The tables of values of U for the forms of structure of each element of these surfaces may give the value of U for them, but if the structure

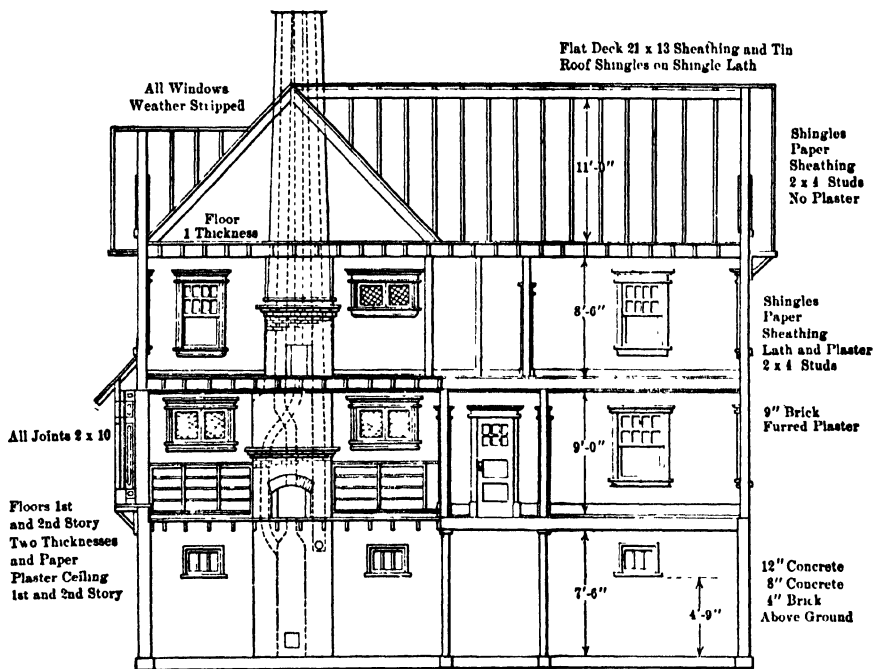


FIG. 94.—Section through Building.

is one for which no tabular value can be found the value of U must be computed as

$$U = \frac{1}{\sum \frac{1}{f} + \sum \frac{L}{k} + \sum \frac{1}{a} + \sum \frac{1}{c}} \text{ or } \frac{1}{\frac{1}{U_o} + \sum \frac{L}{k}} \text{ or } \frac{1}{\frac{1}{U_o} - \frac{1}{f_o} + \frac{1}{f}} \quad (5-1)$$

using the constants for the materials entering the construction. It is to be remembered that following the recommendation of the A.S.H.V.E. a velocity of 15 mph has been used in fixing f_o , and the values of U tabulated are for such values of f_o . If desired, the effect of a different velocity can be made by the formula above.

Air leakage is next computed on the exposed side, using window leakage only as the plastered wall leakage is negligible. Where wind velocity during the cold months is unknown it is taken as 15 mph.

This air leakage should be computed for all rooms, and for a room which is not exposed to the prevailing wind a velocity of 5 mph should be used. The wind is assumed to blow on only one side of any room.

Having these quantities entered, the following expressions are used for heat from each wall element

$$q_t = UA\Delta t \quad (5-2)$$



FIG. 95.—Elevations of Building.

For air leakage, the heat required per cubic foot for one degree rise in temperature is taken as

$$\frac{0.24}{13.35} = \frac{1}{55.6} = 0.018$$

since the air leakage is not definite enough to use more exact methods.

Then

$$q_{al} = 0.018V(t_r - t_o) \quad (5-3)$$

GENERAL DATA

Design No.: 620	Overall Dimensions: 40' X 35' X 40' outline	Heating Season: Sept. 15-May 10	Humidity—Spring.....
Location: Rydal, N. Y.	Cubage: 52,000 cu ft	Degree Days from 65° F: 6800	“ Summer.....
Owner: L. Q. Smith	Heated Volume: 20,310 cu ft	Lowest Temperature: -25° F	“ Autumn:.....
Architect: R. A. Teck	Ratio $\frac{\text{Heated volume}}{\text{Cubage}} = 0.41$	Continued Low Temp.: 0° F	“ Winter.....
Date: Nov. 20, 1935.		Direction of Winter Wind: W	High Summer Temp.....
		Velocity of Wind: 10 mi. per hr	Av. Summer Temp.....

COMPLETE REQUIREMENTS

Room No.	Mbh	Radiator Surface or Requirements	Volume	Ratio	Room No.	Mbh	Radiator Surface or Requirements	Volume	Ratio	Volume	Ratio	Remarks
101	19.7	84.0	3110	37.0								House faces almost North
102	14.3	63.0	2160	34.3								
103	2.1	9.0	400	44.5								
104	6.5	27.0	1390	51.5								
105	10.0	42.0	1620	38.5								
106	12.4	51.0	2110	41.5								
201	15.2	66.2	2910	44.1								
202	11.0	45.0	2000	44.5								
203	3.5	15.0	720	46.5								
204	8.7	35.0	1700	48.5								
205	8.8	36.0	1710	47.5								
206	2.8	12.0	480	40.0								
Total.....	115.0	485.0	20310	42.8	Total							

Fig. 96.—General Data Sheet.

The sun effect with awnings or any other equipment is then considered with summer air conditioning and entered as shown from methods of Chapter III, but for heating no allowance for sun effect should be made as the heating system must be sufficiently large to care for the heat demands on cloudy days. If forced ventilation from a central plant or from a unit ventilator is to be used, this may be noted in space provided on the data sheet although the heat demand in this case includes that for leakage air as leakage occurs even with the plenum system as shown by Rowley.

If amounts of heat from occupants, lights or apparatus in the room are appreciable quantities, these are to be computed and entered upon the sheet. If such quantities are given off at intervals and the room must be held at the desired temperature without these sources of heat, this fact must be noted, as the heating system must supply the total heat losses at times of greatest demand and the heating units are shut off or decreased in effectiveness when the additional heat is supplied from these other sources. If these sources of heat exist whenever the desired temperature is obtained, then the heating system is designed to supply the difference between the heat lost and that developed in the room.

At this point the heat demand for the room is determined by the algebraic sum of the losses and gains, and the heating system is designed to supply this heat. As this heat loss is cared for by different types of equipment, each will be treated in its special chapter, but for each these quantities of the table, Fig. 90, must be computed.

The data sheets for four rooms, Fig. 90, will now be filled out for certain rooms of the building shown in the figures, and a combined sheet for the building, Fig. 96, gives the total demand and general data.

The points of the compass marked on the plans indicate that the building faces slightly east of north, and as the prevailing winds are given as northwest or west, the side of the house at the right hand of the drawing is the exposed side of the building.

The house is in New York State, and so, from the table of Chapter III, -23°F is the lowest temperature. The outside temperature would be taken as $-23^{\circ}\text{F} + 15^{\circ}$ or -8°F according to the recommended practice of the A.S.H.V.E. Guide. In this particular case the records of the town show that 0°F is the lowest temperature to be expected to continue for several days, and for that reason the author has taken t_o as 0° . The weather reports show that winds from the west and northwest during the winter months have a velocity of 10 mph.

The temperature at the breathing line is taken at 70° from Chapter II, and because of the low ceiling heights there will be no correction for mean t_i or the ceiling temperature. t_i for walls and ceiling will then be 70° , and t_i for the floor will be taken as $70^{\circ} - 5^{\circ}$, or 65° . Heat loss from cellar walls below ground and cellar floors will be taken as 2 Btu per sq ft per hr. When the portion of a building which is

wholly or partially below the ground level is fitted up for occupancy or other useful purposes it is known as a basement.

The coefficients of heat transfer, U , are as follows:

8 in. concrete, 4 in. brick for cellar wall above ground.....	0.45
8 in. brick, furring, plaster.....	0.29
Shingles, paper, siding, 2×4 studs, plaster.....	0.20
Shingles, paper, siding, 2×4 studs, no plaster..	0.30
Shingle and paper roof.....	0.49
Wood deck.....	0.49
Partitions.....	0.26
Single glass.....	1.13
Floors—2 thicknesses wood, paper and no plaster (hot room above, first floor).....	0.27
2 thicknesses wood, paper and plaster (mean of 0.03 and 0.16, second floor).....	0.10
plaster and single wood (hot room below)....	0.19

Air Leakage:

Weather-stripped double-hung windows, 15.5 cu ft per hr per ft.....	See rooms
Weather-stripped casement windows, $\frac{3}{4}$ double hung, leakage 10.3 cu ft per hr per ft.....	See rooms
Non-weather-strip swing cellar windows of 21.4 cu ft per hr per ft.....	= 9 ft length
Glass area, attic.....	= 90 sq ft
Walls of attic, net.....	= 420 sq ft
Shingle roof area.....	= 2064 sq ft
Deck area.....	= 273 sq ft
Attic floor.....	= 1500 sq ft
First and second floor area.....	= 1500 sq ft
Cellar wall below ground.....	= 740 sq ft
Cellar wall above ground.....	= 530 sq ft
Cellar floor.....	= 1500 sq ft
Glass cellar windows.....	= 50 sq ft

To find temperature of attic:

$$(90 \times 1.13 + 420 \times 0.30 + 2064 \times 0.49 + 273 \times 0.49)(t_a - 0^\circ) \\ = 1500 \times 0.19(70 - t_a)$$

$$1658t_a = 19,950$$

$$t_a = 12.0^\circ \text{ F}$$

This low temperature is due to the fact that the roof construction of this building had a very high conducting value (poor insulation) and the roof area is large compared with the floor area. Plaster beneath roof would raise the temperature to 21° , while plaster on $\frac{1}{2}$ in. of insulation would raise the temperature to 23° and with rock wool fill in this roof to 32.5° . It is thus seen that except for approximate work the temperature of the attic should not be taken as the mean between 70° and 0° .

For the cellar temperature:

$$(740 + 1500) \times 2 + (530 \times 0.45 + 0.018 \times 21.4 \times 9 + 50 \times 1.13)(t_c - 0)$$

$$= 1500 \times 0.27(65 - t_c)$$

$$703t_c = 21,845$$

$$t_c = 31^\circ \text{ F}$$

All constants being known, the calculations are made as shown in Figs. 90 and 96.

The amounts of heat for each element are computed for each face of the room and added to give the total heat for this face. Windows and doors are considered as openings for which the value of U is taken as 1.13. If the windows are equipped with double glass, then the constant for the windows will be different from that of doors and these must be considered separately.

The air leakage has been computed for 10 mph for north or west sides; had rooms with south or east exposures been placed on the data sheet the leakage at 5 mph would have been used.

As this is a residence the number of occupants of the rooms has not been entered and there are no ventilation requirements.

If any heat is added from the use of the room or from the sunshine during the period for which the heat demand is figured, these quantities are entered in the column so marked and the difference between the heat demand and this so-called occupational heat is the net heat required by the room. Before finally computing the radiation for the room any special allowance, such as that for quick heating of a cold building by adding from 25% to 50%, must be made.

The designer enters the number of square feet of direct radiation for steam or water for such systems, the amount of hot air for furnace or indirect heating, or the number of unit heaters for this type of heating in the column marked "requirements." The actual device used is entered under "install," and finally the ratio of volume to surface is entered, that required and that installed. Space is left for any special notes which are needed to explain results differing from those obtained on other buildings.

On the general data sheet the designer enters all weather information that he can secure for use in heating, ventilation and air conditioning. He then fills out other data for comparisons and obtains the total surface and heated volume for check purposes. The ratio of total heated volume to total heating surface can be used in estimating the probable surface for a projected building as the heated volume is about 70% of the outside cubage or volume of the structure, including basement.

PROBLEMS

1. Secure a set of plans for a small building and mark room numbers on these plans as well as temperatures to be used for t_i in each room. Mark compass points and velocity of wind to be expected for location of building.

2. Make and fill out table for a room with following data: The north wall is 20 ft wide on the exposed side. The east and west walls are partitions 15 ft long. The south partition has a door 4 ft \times 6 ft 8 in. at the center of the length. The room is 11 ft high. It has three occupants at light work. There are two double-hung metal windows in the north wall, each opening being 5 ft \times 8 ft. The wall of 12 in. of brick work is of type *H*, Fig. 35. The partitions are of type *A*, Fig. 42. The floor and ceiling are of type *B*, Fig. 34. The room is to be held at 70° at the breathing line. The hall at the south is at 65°, and the rooms at east and west are at 70° F at the breathing line. The outside temperature is 10° F. The ceiling of room below is at 75° F, while the floor of room above is at 65° F. The wind velocity is 15 mph. How much heat is needed for this room in winter? How much heat must be removed in the summer time if the outside temperature is 95° F and the inside temperature of all parts of the building is at 75° F? Do the 600 watts of electrical energy used for illumination enter into these computations?

3. A convention hall of 1,200,000 cu ft has 20,000 sq ft of walls for which $U = 0.22$; 8000 sq ft of glass in walls and roof and 30,000 sq ft of roof for which $U = 0.25$. The hall seats 3000 persons, and 100 kw are used for illumination. At what temperature should this building be held? What is the temperature beneath roof at 35 ft above floor? At 40 ft above floor? What mean inside temperature would you use for roof? For walls? What would be the heat demand before occupancy? After occupancy? After lights are turned on with occupancy? How much air would you circulate? $t_o = 5^\circ$ F center pivot windows, 5 \times 8.

CHAPTER VI

DIRECT STEAM HEATING

Direct heating refers to the use of individual radiators placed at suitable locations in the rooms to be heated. In direct steam heating and with unit heaters the vapor is carried to the radiators or convectors on the different floors in several ways.

Fig. 97 illustrates the upfeed single-pipe system in which steam flows from the boiler to the main *A* in the basement which pitches $\frac{1}{8}$ in. to

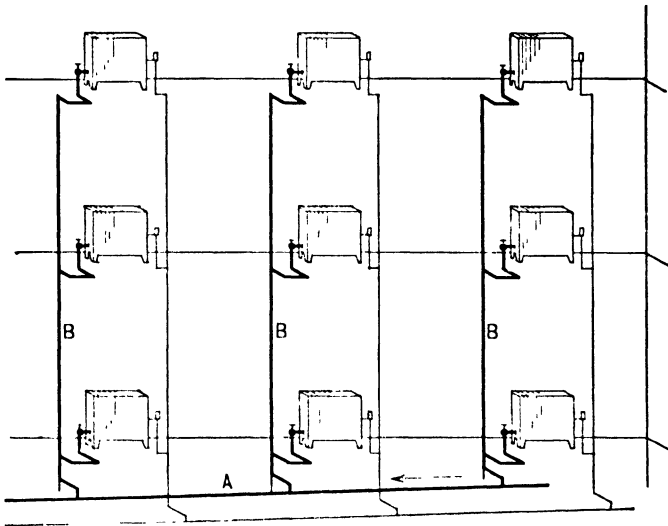


FIG. 97.—Single-pipe System.

$\frac{1}{4}$ in. in 10 ft in the direction of flow from the point connected with the outlet of the boiler or other source of steam (as the supply main of a district heating system) to the point at which the drip or return condensation is carried to the boiler in the gravity return system or to the receiver connected with a return pump in the mechanical return system. The purpose of the pitch in direction of steam flow is to have the condensation travel with the steam, as any attempt to have this water go against the direction of the steam is liable to prevent the flow of the

water and even that of the steam and cause water hammer unless the pipe is 3 in. or more in diameter. Even here, water hammer may occur unless the pipe is dripped (drained) at a number of places along the main. In small plants the low end of the main may be connected to a point below the water line of the boiler, thus returning the condensation and forming a gravity return. Single risers *B* are taken from the top of the main by swiveling branches or runouts to care for expansion, and radiator branch connections or radiator runouts extend from the riser to the single radiator valves connected with the low point of each

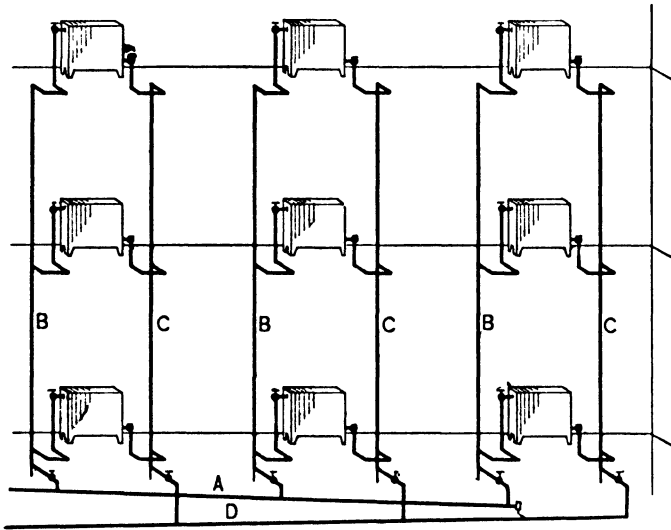


FIG. 98.—Two-pipe System.

radiator. These branches to risers and radiators should be well pitched ($\frac{1}{2}$ in. in 10 ft is recommended by experiment) to prevent retention of water in radiators. The angle valve shown in Fig. 70 must be used with such installations as the same valve must allow condensate to flow back against the steam supply. Each radiator is supplied with an air valve on the side opposite the inlet and at a point below the top of the radiator. These air valves discharge into the room or they may be connected to an air line as shown in the figure. This air line is under low pressure produced by a vacuum pump or an ejector operated by water.

In the upfeed double-pipe system, Fig. 98, the supply main *A* and its steam risers *B* are separate from the return risers *C* and the return main or main drip *D*. The radiator branch connections to the risers are

pitched so that no pockets are formed and expansion is cared for by swiveling at the several pipe fittings used to form the branch. In this system there is no interference between supply and return, and water hammer or noise from mixing will not occur.

By using return valves of the trap form the return line is free from any steam at supply pressure and it is not possible for the discharge from any one radiator to interfere with the discharge from another by building up pressure. These valves will relieve the radiators of air, especially when a vacuum is carried on the return system.

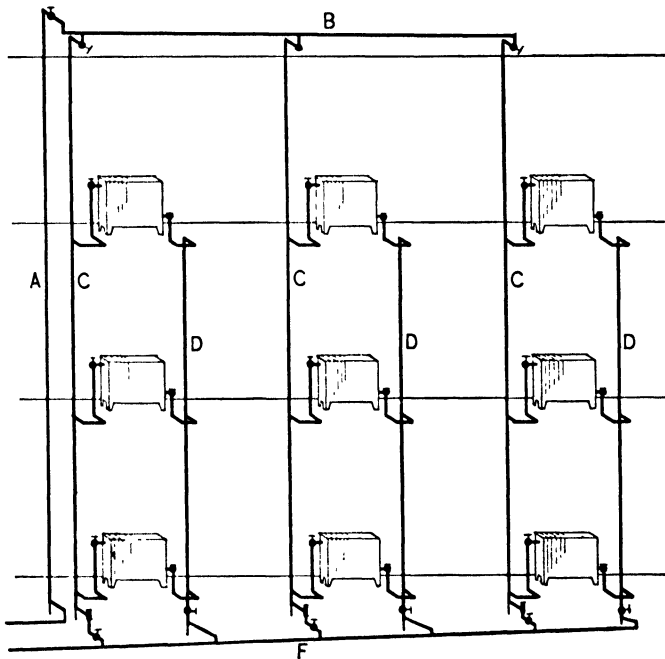


Fig. 99.—Mills or Downfeed Double-pipe System.

Another system of distribution, the Mills or downfeed double-pipe system, is shown in Fig. 99. Steam is carried through the main riser *A* to the distributing main at an upper floor of a building. In high buildings there may be such mains at intervals of 10 to 20 floors, these mains for each zone being carried in a pipe space or gallery, a story of low height, with certain other mechanical equipment. The vertical supply riser may carry high-pressure steam to reduce its size, and this delivers steam through a reducing pressure valve to the distributing main. The main riser *A* may be equipped with steam valve at top or bottom to control admission to the distributing main if more than one riser feeds

this main. The steam risers *C* are connected with the main *B* by valved runouts, and as steam is taken off from the riser to the various radiators its size is reduced until at the bottom of the riser it becomes merely a drip line discharging into the return main *F*, through a return valve of the trap form. The condensate discharged through this valve is produced by the radiation from the riser.

The return risers *D*, which are made larger as they reach lower levels on account of the increase of flow, are finally connected to the return main *F*.

Both risers are connected with their mains by valves so that any one may be cut off from the system when necessary for repair or alteration. To the bottom of each riser an 8-in. or 10-in. nipple is added to the tee for the runout to form a scale trap or dirt pocket which prevents a stoppage in the runout. The cap on the end of the nipple can be taken off to remove scale or other dirt.

Some variations of this system may be used. A main at a tenth floor supplies steam for upflow to the eighteenth floor and for downflow to the first, although in upward distribution through the steam risers there is a certain amount of condensate which flows against the steam and should be removed by a return valve trap at the bottom of this pipe and discharged into a drip line.

If risers supplied from the top are joined to radiators by single connections, this forms a single-pipe downflow system. The return condensation from the radiators is passed back through these connections and is delivered to the return line *F* through a return trap at the bottom of the riser, giving a return main free from steam. This system is sometimes called the complete circuit system.

Whenever return traps are used to drain a line they must be connected with that line by a few feet of uncovered pipe to cool the water by radiation as hot water may prevent the opening of the expansion drip valve. With a float-controlled drip trap, Fig. 100, however, this bare pipe is unnecessary. The inlet is at *C*, and on lifting the ball float by the condensate the liquid flows past the valve *B*. Air is discharged by the expansion bellows *D* through the pin valve as long as steam is not present. This prevents the trap from becoming air bound.

The return main *F* is connected with a receiver when the mechanical system of return is used, this receiver emptying into a boiler feed pump, a vacuum return pump or a return trap. In the gravity return the pipe *F* is connected with the boiler at a point below the water line, as shown in Fig. 6. If the return main of the gravity system is placed below the water line of the boiler it is called a wet return; return mains above this level are called dry returns. Air must be allowed to escape

from the return, and when dry, special air relieving devices are used at the point of connection of this dry return to the low point on the boiler. Whenever a line contains steam and water it is possible to prevent the entrance of steam into the drain from this line without the use of a trap valve by discharging the water into a water seal, Fig. 101, consisting of two pipe legs of sufficient length to care for the difference in pressure between that of steam in the line which is to be drained and that of the drip line.

Systems constructed to operate at pressures very near that of the atmosphere and even at times below the atmosphere are known as vapor systems.

Concealed Piping. In all these systems the main risers are often placed in small spaces formed by partitions called pipe shafts. Repairs

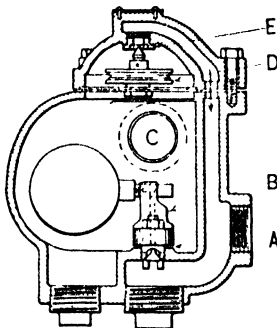


FIG. 100.—Webster Drip Trap.

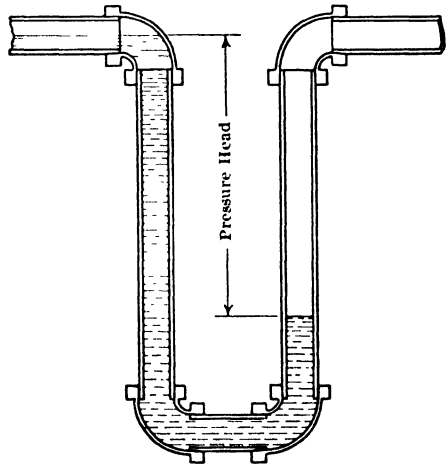


FIG. 101.—Water Seal.

are rare on such lines, but it is always well to build the shafts with doors at intervals and possibly with one long removable panel for replacements. In all design work, thought should be given to the possible method of making repairs or replacements.

Radiator risers may be concealed in chases formed in the walls whenever concealed piping is desired. The author usually prefers to run these and the radiator branches exposed, as then repairs can be made without cutting walls or plaster. Exposed pipes are not objectionable to some users.

When branches are concealed they are placed in the space between the floor slab and the hung ceiling before the latter is formed, or put in a chase above the floor slab. In many installations distributing mains

and other service lines are carried above the hung ceilings of the hallways. This space between the ceiling of the corridor and the floor slab is frequently used to carry the distributing ducts for conditioned air supply.

Radiator Requirements. To find the total square feet of radiating surface required for a room for steam radiation the heat demand for the room per hour, determined as in Chapter V, is divided by the transmission per square foot of radiation per hour. This is 240 Btu for standard temperature conditions, but for different room or steam temperatures this number is changed by the use of the factor given in Table A-9, page 404. Wall radiation of cast iron or of pipe coils gives transmissions differing from 240 Btu per hr per sq ft as mentioned in Chapter IV.*

This total surface found for any room is then considered in connection with the possible desirable positions for radiators or convectors, together with the space limitations of each position. As stated earlier, the radiators are usually placed beneath windows, as this space is not used for permanent furniture and warm air currents rising from the radiator will blanket the cold air from the window, distributing the air more uniformly throughout the room. Table A-6 indicates the space requirements for radiators of different sizes, and this information will enable the designer to select the sizes for the positions which he desires to use to obtain uniform heating. With convectors and other radiators of like form, the manufacturers' tables give the value of the radiators of definite size in terms of equivalent direct radiation as soon as the designer fixes the height of stack formed behind the panel for the convector or fixes the air discharge from the unit heater.

In selecting radiators it must be realized that a long, thin radiator of small height will distribute heat more uniformly throughout a room and cause less variation in temperature at various heights in the room than any other form, and where possible such should be used. Tests made by Kratz and Fahnestock have shown that with thick radiators (5 tube, $23\frac{1}{2}$ in. high, $21\frac{1}{2}$ in. long) the variation of temperature between floor and a 9-ft ceiling was 21.6°F ; with thinner radiators (3 tube, $23\frac{1}{2}$ in. high, 28 in. long), 17.5°F ; with Fantom radiators (17 in. high and 40 in. long), 13.8°F , and with wall radiators ($13\frac{5}{8}$ in. high, $51\frac{5}{8}$ in. effective length), 10.8°F . The lower temperatures of the ceiling with the longer radiators and a general lower average temperature resulted in less heat loss from the room and hence less heat required from the radiator.

* A new method of rating radiators suggested by the A.S.H.V.E is to give the number of thousands of Btu per hour (written Mbh) transmitted by a radiator in place of 240 Btu per sq ft per hr under standard conditions. In this case, however, the method of procedure would be the same as that outlined.

The concealed radiators deliver practically all their heat by convection, and when the hot air is delivered from the front face they give a more uniform temperature throughout the room than that obtained with the open radiators, being very efficient as they cut down the heat loss from the room by giving a lower average room temperature over the wall.

The variation of temperature at different elevations in a room is indicated by a curve, Fig. 102, from the laboratory of the Crane Company showing the distribution from a 32-in. 5-tube radiator and from their concealed radiator in a 32-in. enclosure. The figure shows that with the exposed radiator the temperature is 70° F at the breathing

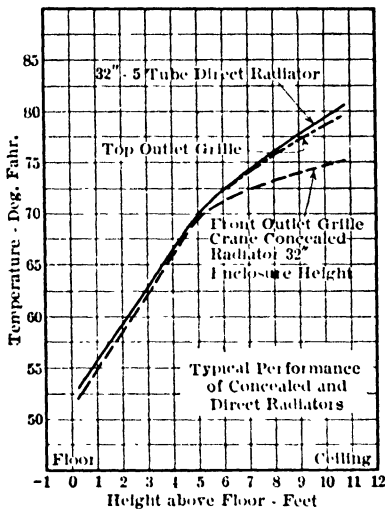


FIG. 102.—Temperature Variation in Tests by Crane Co.

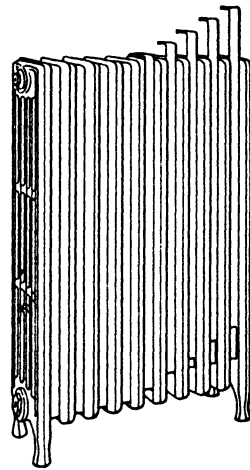


FIG. 103.—Crane Radiator and Invisible Shields.

line, 60 in. above floor, 55° F at the floor, 62° F at 30 in., the knee or comfort line, and 79° F at the 10-ft ceiling height, whereas with the concealed radiator the temperature is 74° F at the ceiling with the temperature at the other heights slightly lower than those with the direct radiator. When the grille is moved from the front of the enclosure to the top, the temperature at elevation 10 ft is nearly as much as that for the direct radiator. The outlet from a front grille will direct more of the heated air from the concealed radiator into the room, giving better distribution and a lower temperature at the ceiling. It must be remembered in considering this figure that a radiator narrower than 5 tubes would have given a lower ceiling temperature.

The deflection of the air into the room from a direct radiator may be accomplished by the use of the Crane invisible shield, Fig. 103. These

metal strips are placed between sections at the back of the radiator and have a right-angle member at their upper end extending half way across the radiator, deflecting the air out into the room. These reduce the ceiling temperatures and raise the temperature below the breathing line. When radiators are placed against the wall the invisible shields reduce the dust current which is found on the walls behind radiators. Sheet-metal shields also are placed behind and above radiators for this purpose.

It must be remembered that an enclosure around a radiator will give less direct heat into the wall behind it, thus, decreasing the wall loss.

In computing the amount of radiation for a given room, reference is made to the allowances which should be made for intermittent heating or heating at intervals. If sufficient time is available for heating after a long shutdown, no allowance may be necessary, but for quick heating the surface and its piping should be increased 50%.

The amount of radiation needed in the direct heating system for the building illustrated in Chapter V has been determined by dividing the net heat demand per hour by 240 Btu and then dividing this radiation between suitable locations. The assumption of 70° room temperature and 215° F steam is the reason for the use of the number 240. Actual radiator sizes have then been selected and entered on the data sheet, being finally transferred to the general data sheet, Fig. 96.

Volume Ratio. Having computed the total radiation for a given room, the author computes the volume ratio by dividing the volume of the room by the square feet of heating surface. This gives the cubic feet cared for by 1 sq ft of radiation and is compared with past experience as a check. Any appreciable variations from averages obtained previously must be accounted for by the peculiar features of the room, or some error must be present in the calculations. As a guide the following values are given:

TABLE 6-1

CUBIC FEET OF VOLUME PER SQUARE FOOT OF DIRECT STEAM RADIATION

Residences: Living rooms.....	35 to 60
Sleeping rooms.....	50 to 80
Offices.....	40 to 80
Schools.....	40 to 80
Factories.....	75 to 100
Assembly halls.....	75 to 100
Hotels.....	75 to 100
Stores.....	75 to 100
Churches.....	125 to 200
Auditoriums.....	125 to 200
Gymnasiums.....	100
Work shops.....	150

Steam Lines. After determining the type of system used for the proposed design, the next step in design is to arrange the branches or runouts from the radiators to risers using Table 6-6 on page 166 to determine the sizes of runout for each radiator. The manufacturers of radiators have standard tapplings such as those given in Table 6-2, but frequently makers of modulating valves and return trap valves use smaller sizes in two-pipe systems necessitating bushings in the manufacturers' tapplings.

TABLE 6-2
TAPPINGS FOR RADIATORS IN INCHES

Single Pipe		Two-pipe Steam			Hot Water			Vapor		
Sq Ft Area	Tap- ping	Sq Ft Area	Sup- ply	Re- turn	Sq Ft Area	Sup- ply	Re- turn	Sq Ft Area	Sup- ply	Re- turn
24	1	48	1	$\frac{3}{4}$	40	1	1	30	$\frac{1}{2}$	$\frac{1}{2}$
24-60	$1\frac{1}{4}$	48-96	$1\frac{1}{4}$	1	40-72	$1\frac{1}{4}$	$1\frac{1}{4}$	30-60	$\frac{3}{4}$	$\frac{1}{2}$
60-100	$1\frac{1}{2}$	96	$1\frac{1}{2}$	$1\frac{1}{4}$	72	$1\frac{1}{2}$	$1\frac{1}{2}$	60-125	1	$\frac{1}{2}$
100	2	125	$1\frac{1}{4}$	$\frac{1}{2}$

TABLE 6-3
BARNES AND JONES VALVE SIZES IN INCHES
(Two-pipe System)

Modulation		Thermostatic Traps		
Sq Ft Area	Size	Sq Ft Area	Inlet	Outlet
30	$\frac{1}{2}$	200	$\frac{1}{2}$	$\frac{1}{2}$ or $\frac{3}{4}$
60	$\frac{3}{4}$	400	$\frac{1}{2}$ or $\frac{3}{4}$	$\frac{3}{4}$
100	1	700	$\frac{3}{4}$	$\frac{3}{4}$
180	$1\frac{1}{4}$	1200	1	1

The thermostatic traps above are figured on $\frac{1}{4}$ lb steam per hr per equivalent sq ft of direct radiation and a $1\frac{1}{2}$ lb differential between steam and drip, and for vapor systems of from 4 to 8 oz of differential pressure the capacities are reduced to one-half of the values in the tables.

The branch connections from the radiator to the risers should be

arranged so that as the risers expand and contract, the radiators will not be lifted from the floor, and they must be pitched $\frac{1}{2}$ in. in 10 ft to the riser so that condensation will drain. The best way to arrange these branches for a single-pipe system is shown in Fig. 104, an oblique drawing. The oblique method of representing piping is very useful. In this the vertical lines represent vertical pipes; horizontal lines, those parallel to the plane of the paper; and lines at 30° to the horizontal represent those perpendicular to the plane of the paper. At times two 30° lines and a vertical line are used for the three directions. Actual distances are measured in the three directions. Single lines serve to represent the pipes, and these may be solid, dotted or dot and dash lines. Thus, Fig. 105 represents four ways of connecting radiators. The third method should never be used, as the expansion of the riser will lift the radiator, tending to break the fittings if it does not actually break them.

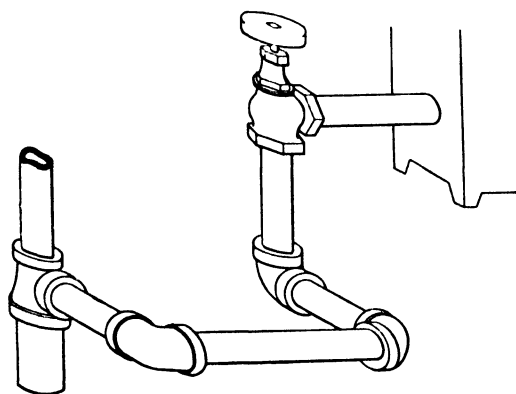


FIG. 104.—Radiator Runout or Branch.

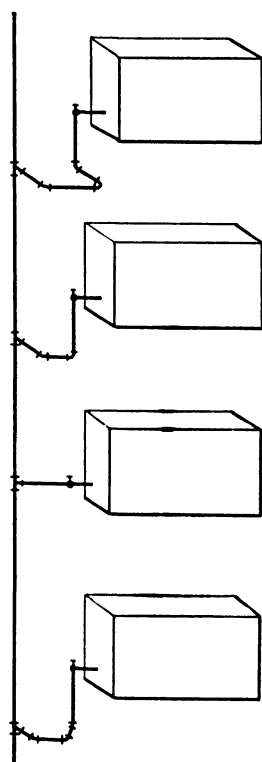


FIG. 105.
Radiator Branches or Connections.

The second method is satisfactory if the branch is long enough to have some spring, and in the first or best method, the branch will swivel on the ells and care for the expansion.

Although these figures show connections for a single-pipe system, the views might represent the branch connections of the drips or returns of a two-pipe system, and by extending the vertical line to the top of the radiator it would represent the supply. Regular cast fittings are used with full-weight pipe.

Risers and Mains. Risers are usually arranged so that a number of radiators come on the same line. This cuts down the number of risers and gives a less expensive installation. To study the distribution properly it is well to make a small-scale drawing, usually $\frac{1}{8}$ in. to the foot, showing the development of the walls of the building from the inside, with windows and partitions marked on it, and on this the radiators and risers are placed after using plans to determine preferable locations.



FIG. 106.—Development of Building.

Fig. 106 shows such a development for the building figured in Chapter V. On this view the radiators are placed and each radiator and riser is given a designating mark, the number of the room with a letter to denote the particular radiator of this room. Thus in room 105 there are two radiators, 105a and 105b. The risers are marked A, B, C, D and E, or by some other symbols. The radiators and risers are finally drawn in and marked on the plans. Care must be taken in placing these, when exposed, to see that there is no interference with the location of the

p = the pressure drop in L feet in pounds per square inch;
 W = the weight of steam in pounds per minute;
 d = inside diameter of pipe in inches;
 D = density of steam at mean pressure in pounds per cubic foot;
 L = length of pipe in feet in which pressure drop of p pounds per square inch occurs.

The pressure drop in steam systems is very slight so that it may be given in ounces per square inch per 100 ft or inches or feet of water per 100 ft or in decimals of a pound per sq in. per 100 ft of length. The relation between these is given below:

0.1 lb per sq in. = 1.6 oz per sq in. = 0.23 ft water = 2.77 in. of water
 0.0625 lb per sq in. = 1.0 oz per sq in. = 0.14 ft water = 1.73 in. of water.
 0.036 lb per sq in. = 0.58 oz per sq in. = 0.083 ft water = 1.0 in. water.

The discharge of steam through a pipe depends on the drop in pressure per foot of length of pipe, and to determine the equivalent length of pipe, the resistance brought on by the fittings must be cared for in addition to the length of straight pipe. The resistances of elbows, valves and other fittings have been experimentally determined as equal to the resistance of a certain length of pipe of the same diameter which has the same pressure drop. The lengths may be expressed as feet of pipe or number of diameters, the first being specific, the second a general method. The A.S.H.V.E. Guide for 1936 gives the following equivalent lengths of valves and fittings.

TABLE 6-4

LENGTH OF PIPE IN FEET HAVING SAME RESISTANCE AS FITTING

(From the 1936 Guide of the A.S.H.V.E. Printed by permission.)

Size Pipe	Standard Elbow	Side Outlet Tee	Gate Valve	Globe Valve	Angle Valve
2	5	16	2	18	9
2½	7	20	3	25	12
3	10	26	3	33	16
4	14	35	5	45	22
5	18	44	7	57	28
6	22	50	9	70	32
8	31	63	12	94	42
10	39	76	15	118	52
12	47	90	18	140	63
14	53	105	20	160	72

To reduce the resistance of the pipes at couplings and fittings or even at welded joints, the ends of the pipes should be reamed out after cutting to length, and any pipe joint cements or compounds used on the threads should not be allowed to extend over the end. Burrs left on the pipe have been shown by Houghten and Ebin to reduce the capacity of a given pipe 25%, and even a square unreamed end may cut down the flow 10%. The inside diameter of pipe varies from its listed size so that the carrying capacity may be less than that which is estimated.

The amount of steam condensed per hour per square foot of E.D.R. (Equivalent direct radiation) under standard conditions to give 240 Btu is so near $\frac{1}{4}$ lb per hr that in all heating problems this value is used as the condensation per square foot per hour, so that the pounds per minute of (6-1) may be changed into square feet of equivalent direct radiation by multiplying weight of steam per minute by 4×60 .

This formula is now applied to determine the surface A to be cared for by a pipe of diameter d with a drop of $\frac{1}{4}$ lb per 100 ft with steam at 215° F.

$$\begin{aligned} A^2 &= (4 \times 60 \times W)^2 = \frac{0.03942 \times d^5 \times 16 \times 3600}{400 \times 0.000131 \times \left(1 + \frac{3.6}{d}\right)} \\ &= 433 \times 10^4 \frac{d^5}{1 + \frac{3.6}{d}} \end{aligned} \quad (6-2)$$

For a 2-in. pipe of inside diameter 2.07 in., the fraction becomes 13.83 and $A^2 = 599,000$.

$$240W = A = 773 \text{ sq ft}$$

From this value, the area A cared for by any other diameter d would be found by

$$240W_d = A_d = \frac{773}{\sqrt{13.83}} \left(\frac{d^5}{1 + \frac{3.6}{d}} \right)^{\frac{1}{2}} = 207.5 \left(\frac{d^5}{1 + \frac{3.6}{d}} \right)^{\frac{1}{2}} \quad (6-3)$$

Since the quantity W or the area A varies as $(p/L)^{\frac{1}{2}}$ the area at some other pressure drop per foot, p/L or p per 100 ft, would be given by

$$A_{\frac{p}{L}} = 20A_{\frac{1}{100}} \sqrt{\frac{p}{L}}$$

or

$$A_p = 2A_{\frac{1}{100}} \sqrt{p} \quad (6-4)$$

The variation of A with density is given by

$$A_p = A_{215^\circ} \sqrt{\frac{D}{0.03942}} = 5.03 A_{215^\circ} \sqrt{D} \quad (6-5)$$

These formulae may be used for any lengths, pressure drops, sizes or pressures. They can be used for power-house work as well as for heating problems. Table 6-5 gives the square feet of E.D.R. determined by the formulae for different sizes of pipe with different pressure drops per 100 ft when the average steam condition is 215° F and each square foot uses $\frac{1}{4}$ lb of steam per hr.

TABLE 6-5

EQUIVALENT DIRECT RADIATING SURFACE CARED FOR BY SUPPLY MAINS AND RISERS

For steam at 215° F. One- or two-pipe systems. $\frac{1}{4}$ lb of steam per sq ft per hr

Nominal Pipe Size	Steam and Condensate Flow in Same Direction					Upfeed 1 oz per 100 ft †
	Pressure Drop in 100 ft					
	0 05%	0 10%	0 15%	0 25%	0.50%	
1	50	71	86	112	158	56
1¼	110	156	190	246	348	123
1½	170	241	295	381	539	191
2	346	489	597	773	1 092	380
2½	566	800	980	1 266	1 788	635
3	1 038	1 470	1 800	2 320	3 280	1160
4	2 205	3 120	3 820	4 930	6 970	2490
5	4 020	5 680	6 970	9 000	12 040	4550
6	6 710	9 490	11 640	15 000	21 200	7530
8	14 550	20 550	25 200	32 500	46 000	
10	25 300	36 700	43 700	56 500	79 800	
12	41 400	58 500	71 700	92 500	129 200	
14*	51 500	72 800	89 100	115 000	162 500	
16*	73 800	104 500	128 000	165 000	233 000	

* O. D. Pipe.

† From test at A.S.H.V.E. Laboratory.

The fact that water of condensation has to be carried in steam lines makes it necessary that the effect of this water be determined by experiment. Many excellent experiments have been made at the Research Laboratory of the A.S.H.V.E. at Pittsburgh, associated with the station of the Bureau of Mines, by Houghten, Ebin, Lincoln, O'Connel and

Gutberlet at various times from 1922 to 1931 and reported in the Transactions of the A.S.H.V.E. The experiments were made on mains, risers, branches, runouts, and fittings for one- and two-pipe systems, and the tables giving pipe capacities for upflow risers, return risers and mains, as well as runouts, are based on their conclusions and have been compared with tables given in the Guide of 1936.

When steam and the condensate travel in the same direction the water occupies such a small amount of the volume that no allowance has been made in the first five columns of Table 6-5. When water flows against the steam, the experimental results are used for the last column, giving the E.D.R. cared for by vertical pipes. When these are used for supply and return as in the one-pipe system, the next larger size of pipe is to be used.

The E.D.R. cared for by different sizes of runouts, short branches and radiator connections with steam at 215° F, pressure drop about 0.015% and a pitch of $\frac{1}{2}$ in. in 10 ft is given in Table 6-6, using the experiments mentioned above.

TABLE 6-6

E.D.R. FOR RUNOUT SIZES FOR BRANCHES TO RADIATORS

Nominal Pipe Size	One-pipe System		Two-pipe System	
	5 ft Run	10 ft Run	5 ft Run	10 ft Run
$\frac{3}{4}$	27		29	
1	30	26	35	33
$1\frac{1}{4}$	66	60	77	72
$1\frac{1}{2}$	99	89	113	104
2	189	164	204	184

The sizes of return mains are selected from Table 6-7, which has been computed from the results of Houghten and Gutberlet on a 1-in. main, clearing the radiators of air in 40 min.

For return risers with air present, the experiments indicate capacities greater than those given for dry return mains, but on account of the co-action of the return riser and the return main, following the apparent practice in the 1935 Guide, the values given for the return mains under the dry condition will be used as the capacity of the risers. The wet mains have been computed from the pressure drops, assuming the diameters to be reduced to 81% of their nominal size for safety.

TABLE 6-7

EQUIVALENT DIRECT RADIATION CARED FOR BY RETURN MAINS

Water at 60 lb per cu ft

Nom- inal Pipe Size	Pressure Drop in 100 Ft and Pitch of $\frac{1}{4}$ In per 100 Ft									
	0.025%		0.05%		0.10%		0.15%		0.25%	
	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet	Dry	Wet
$\frac{3}{4}$	130	190	145	270	161	390	189	480	226	620
1	268	390	298	550	340	780	388	960	463	1 240
$1\frac{1}{4}$	590	730	650	1 030	725	1 460	850	1 780	1 020	2 300
$1\frac{1}{2}$	910	1 440	1 010	2 030	1 120	2 880	1 320	3 520	1 570	4 550
2	1 810	2 130	2 070	3 000	2 290	4 260	2 700	5 220	3 220	6 720
$2\frac{1}{2}$	3 040	3 960	3 380	5 600	3 740	7 920	4 400	9 700	5 250	12 520
3	5 560	6 150	6 180	8 680	6 850	12 300	8 040	15 100	9 600	19 400
4	11 900	10 600	13 300	14 500	14 700	21 200	17 300	26 100	20 600	33 600
5	21 700	21 100	24 200	29 700	26 800	42 200	31 500	51 600	37 500	66 600
6	36 100	37 200	40 100	52 500	44 500	74 400	52 300	91 000	62 500	117 600
8	74 700	58 500	82 200	82 700	92 200	117 000	108 000	143 500	129 000	185 000

For dry return risers use size of dry return main.

For wet return risers use one size smaller than wet return main

The risers, if really wet, may be taken one size smaller than the wet return main required for the same surface.

If these tables are now applied to a building the riser sheet gives the total radiation on each riser for the selection of the sizes of its parts, and if the amounts of surface supplied by each riser be totaled toward the boiler or supply point in a building of a district system the sizes of the mains are determined.

Design for the Building of Chapter V. The cellar plan, Fig. 108, is added so that the elbows in the various circuits may be counted for the selection of sizes.

Radiator Connections. In this two-pipe system, Fig. 107, the radiators 103a, 104a, 105a, 105b, 203a, 204a, 204b, 205a, 205b and 206a will have $\frac{3}{4}$ in. connections as the sizes (Table 6-6) are all less than 29 sq ft. For radiators 102a, 102b, 201a, 201b, the branch will be 1 in., and for radiators 101a, 101b, 106a and 202a a $1\frac{1}{4}$ in. branch will be used. The drop in these connections is very slight, about 0.015%.

Risers and Mains. In determining the sizes of risers and mains the total drop in pressure in the longest line must be assumed and then the drop per 100 ft. The longest line is that of riser G. The plans show that the length in the steam main from the boiler to G is 84 ft with 20 elbows, and if a total drop of 0.50% is assumed, which is approximately 0.10% per 100 ft, this requires a 2 in. pipe for 344 sq ft E.D.R., the first approximation. The elbows are equivalent to 100 ft of pipe. The

total equivalent length is then 268 ft, so that the assumed drop of 0.10% per 100 ft can be used.

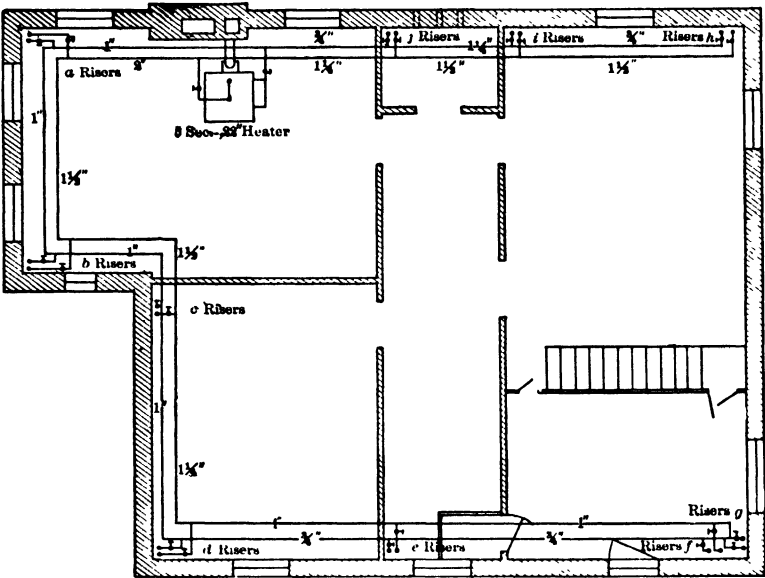


FIG. 108.—Cellar Plan with Boiler and Mains.

From Tables 6-5 and 6-7, the riser sizes are selected as follows:

TABLE 6-8
RISER SIZES IN INCHES (Tables 6-5 and 6-7)

Number	Supply				Return
	Upper Part		Lower Part		
	E.D.R.	Size	E.D.R.	Size	Size
A	33.0	1	75 0	1 $\frac{1}{4}$	3 $\frac{3}{4}$
B	33 0	1	75.0	1 $\frac{1}{4}$	3 $\frac{3}{4}$
C	31 5	1	3 $\frac{3}{4}$
D	45.0	1	76.5	1 $\frac{1}{4}$	3 $\frac{3}{4}$
E	15.0	1	24.0	1	3 $\frac{3}{4}$
F	17.5	1	44.5	1	3 $\frac{3}{4}$
G	17.5	1	17.5	1	3 $\frac{3}{4}$
H	18.0	1	39.0	1	3 $\frac{3}{4}$
I	18.0	1	39.0	1	3 $\frac{3}{4}$
J	12.0	1	63.0	1	3 $\frac{3}{4}$

For the steam and return mains the following sizes will be used:

TABLE 6-9
SIZES OF MAINS

Portion Between	Supply		Return
	E.D.R.	Size	Size
<i>GF</i>	17 5	1	$\frac{3}{4}$
<i>FE</i>	62.0	1	$\frac{3}{4}$
<i>ED</i>	86 5	$1\frac{1}{4}$	$\frac{3}{4}$
<i>DC</i>	162 8	$1\frac{1}{4}$	1
<i>CB</i>	194 0	$1\frac{1}{2}$	1
<i>BA</i>	269.0	$1\frac{1}{2}$	1
<i>A Blr.</i>	344 0	2	1
<i>HI</i>	39 0	1	$\frac{3}{4}$
<i>IJ</i>	78.0	$1\frac{1}{4}$	$\frac{3}{4}$
<i>J Blr.</i>	141 0	$1\frac{1}{4}$	$\frac{3}{4}$
Boiler Size	485 0	2	$1\frac{1}{4}$

In such small installations it may be preferable to make the risers of one size throughout, *A*, *B*, *D* and *J* being $1\frac{1}{4}$ in., and for the steam mains $1\frac{1}{2}$ in. pipe may be used for the full run from *A* to *G* and *J* to *H* with a 2 in. line from the boiler to *A* and *J*. The return line may be made 1 in. throughout.

It is advisable to use nothing smaller than 1 in. pipes for steam lines as the corrosion which may occur reduces the area of smaller pipes so much that flow is diminished.

Large Riser. To further illustrate the use of the pipe size tables, an application will be made to the two downflow risers caring for radiators of ten stories of a building, the radiation from each floor being given in the table. It is assumed that a drop of 0.05% per 100 ft with dry return is desired. The capacity at each floor is found by adding the radiation from the bottom upward to care for a downfeed steam riser and from the top downward to care for the downfeed return riser.

TABLE 6-10
RISER XX

Floor	Radiation	Sum from Top	Size Return	Sum from Bottom	Size Supply
10	75	75	1185	4 (3?)
9	100	175	$\frac{3}{4}$	1110	4 (3?)
8	120	295	1	1010	3
7	100	395	1	890	3
6	80	475	$1\frac{1}{4}$	790	3
5	60	535	$1\frac{1}{4}$	710	3
4	200	735	$1\frac{1}{4}$	650	3
3	150	885	$1\frac{1}{2}$	450	$2\frac{1}{2}$
2	885	$1\frac{1}{2}$	300	2
1	300	1185	$1\frac{1}{2}$	300	2
Basement	1185	2	1

The author now prepares a radiator sheet for his specifications. This is of great value to bidders and contractors and permits more definite estimates of costs.

TABLE 6-11
RADIATOR LIST FOR SPECIFICATIONS
FIRST FLOOR

Room	Radiator	Heating Surface		Temperature	Ratio
101	101a 5 tube 26"	42		70	37
	101b 5 tube 26"	42	84		
		—			
102	102a 5 tube 26"	31.5		70	34.3
	102b 5 tube 26"	31.5	63		
		—			
103	103a wall	9	9	70	44.5
104	104a 3 tube 32"	—		70	51.5
		27	27		
		—			
105	105a 5 tube 26"	21		70	38.5
	105b 5 tube 26"	21	42		
		—			
106	106a 4 tube 38"	51	51	70	41.5
		—			
	Total . . .	276			

SECOND FLOOR

201	201a 5 tube 23"	33		70	44
	201b 5 tube 23"	33	66		
		—			
202	202a 5 tube 23"	45	45	70	44.5
203	203a 3 tube 32"	—		70	46.5
		15.0	15		
		—			
204	204a 4 tube 23"	17.5		70	48.5
	204b 4 tube 23"	17.5	35		
		—			
205	205a 5 tube 23"	18		70	47.5
	205b 5 tube 23"	18	36		
		—			
206	206a 3 tube 23"	12	12	70	40.0
		—			
	Total . . .	209			
	Grand Total	485			42.8

The development of a gymnasium, Fig. 109, illustrates the radiators placed on the various risers *A* to *S*. Each radiator is marked with its

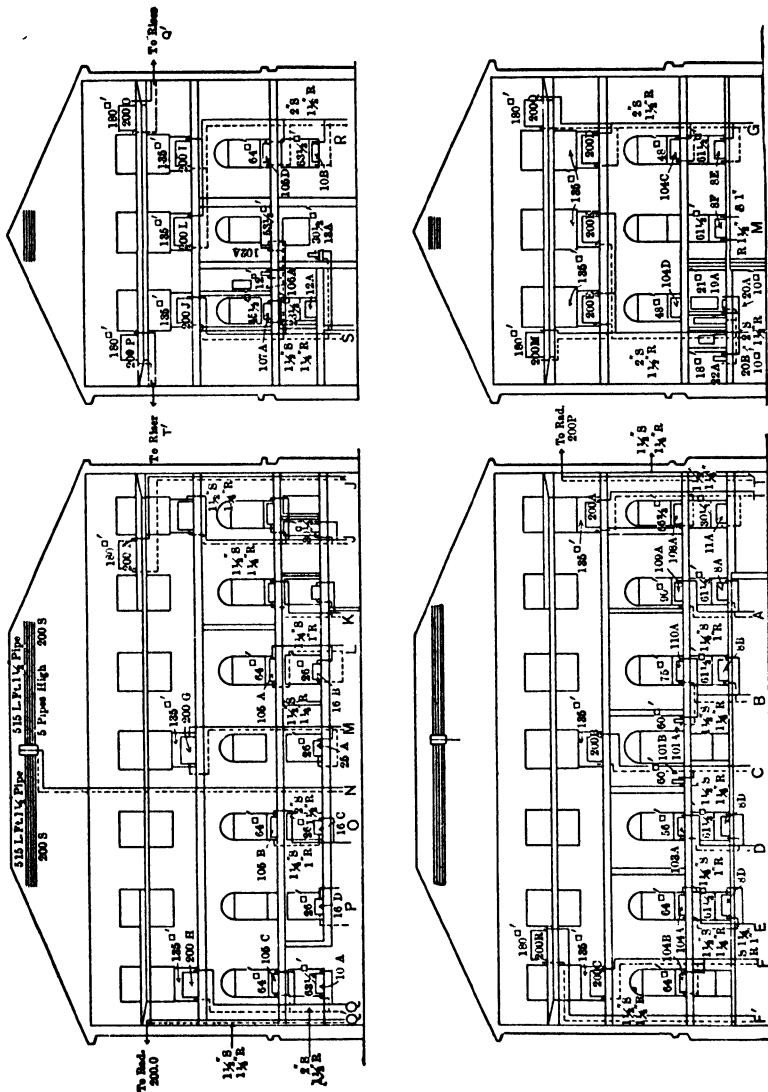


Fig. 109.—Development of a Gymnasium.

room number and letter as well as the number of square feet of equivalent direct radiation.

Floor and ceiling plates, Fig. 110, are placed around the pipes where risers pass through floors or where horizontal pipes pass through parti-

tions for the purpose of closing the end of the hole in the plaster or floor and of giving a neat finish. These are sometimes attached to the ends of the sleeves used to surround pipes which pass through floors and partitions. The sleeves, Fig. 111, are made of cast iron or galvanized iron.



FIG. 110.—Floor and Ceiling Plates.

They form air spaces around the pipes and also allow the pipes to expand and contract without breaking the plaster. By the use of an air space around the pipe the chance for charring timber construction is lessened and the resulting fire hazard is removed.

Risers are supported by pipe hangers, Fig. 112, attached to a top

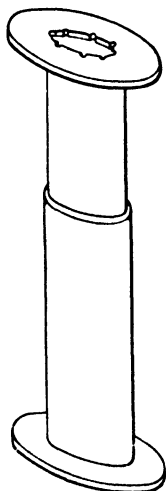


FIG. 111.—Pipe Sleeve.

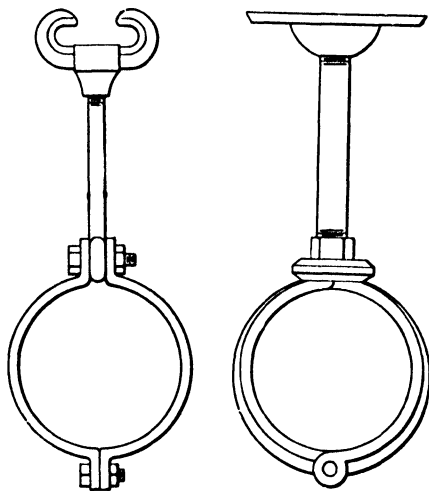


FIG. 112.—Pipe Hangers.

horizontal runout of the riser or by band anchors, Fig. 113, attached around the vertical line and supported at a selected floor line. The anchor type is the better one to employ, as this supports the riser near the base or at the middle of its length as desired. If it is placed at the middle, expansion occurs in each direction. If the end of a riser has

any expansion the runout from the feed line or return must be made with elbows to allow for this. Of course it is possible for long pipes to bend sufficiently to permit expansion, but as there is danger of breaking the fitting it is well to arrange the connection from main to riser as in *A*, Fig. 114. This connection, because of the slight movement possible at the threads of the elbows, allows for the vertical movement of the riser and the horizontal movement of the main due to expansion. If the end of the vertical pipe is anchored near the point of connection, the runout may be made as at *B*, Fig. 114, which allows for expansion of the main. The connections *A* and *B*, Fig. 114 permit the main to move without danger

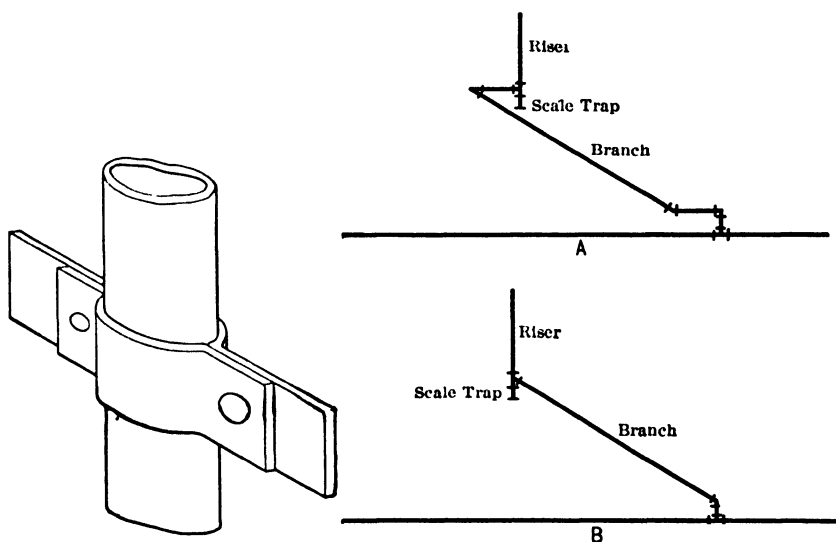
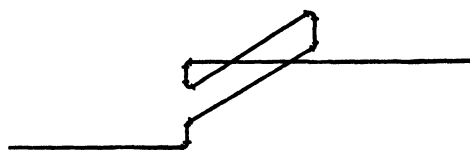


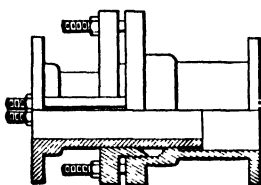
FIG. 113.—Pipe Clamp or Anchor. FIG. 114.—Branch Connections or Runouts.

of rupture, as the vertical elbows are used for this purpose. Expansion in the pipes of most heating systems can be cared for by swing ells, and when possible this should be done. Swing ells, *A*, Fig. 115, slip expansion joints, *B*, expansion bends, *C*, corrugated pipes, *D*, and variators, *E*, are all used to care for expansion. If the slip expansion joint is used it should be so placed that it may be packed without difficulty and the installation made so that repacking may be done without great interference with operation.

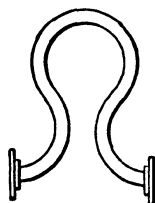
The amount of expansion (which always occurs) varies with the steam pressure. The table on page 175 gives the amounts to be cared for in inches for each 100 ft with different original temperatures and with different steam pressures.



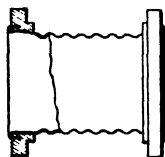
A



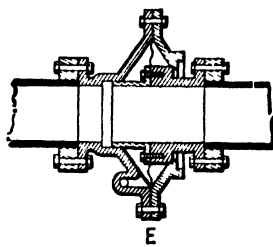
B



C



D



E

FIG. 115.—Expansion Devices.

TABLE 6-12
EXPANSION IN INCHES
Wrought Iron and Steel. Per 100 feet

Original Temperature	Gage Steam Pressure					Temperature	
	150%	100%	40%	10%	5%	212° F	180° F
0° F	3 07	2.84	2 41	2 02	1.92	1 78	1 51
30	2 82	2 58	2.16	1 77	1 66	1 53	1 26
60	2.57	2 23	1.91	1 51	1 41	1.28	1.01
90	2 32	2.08	1.65	1 26	1 16	1.03	0.76

The riser runouts should be made so that no pockets are formed, the lower ends being connected as shown in Fig. 114, whether these be steam or return risers. If the riser itself has a drip extending below its point of attachment, this branch connection could be made from the lower side of the main *A*. The runout is taken from the top of a distributing main if dry steam is desired, the condensation in the main being dripped at some low point. If it is desired to care for drip from the main by the risers, then the connection is made to the bottom side of the main.

Mains. The methods of support and expansion are those described for risers, and the hangers are placed at distances sufficiently short to prevent sagging. Ten to fifteen feet may be used for the distances between supports. Drainage should be cared for by pitching the line in direction of flow, to prevent stoppage and noise, by giving $\frac{1}{8}$ in. to $\frac{1}{4}$ in. to 10 ft. When, however, this is not possible throughout the length

of the pipe because of the building layout, the pipes should be run with the proper pitch as far as possible and then by means of a riser and four ells, Fig. 116, the line is brought to a higher level for continued pitching in direction of flow, the drip at the low point formed at this riser being cared for by a trap. When pipes of 3 in. and above are used

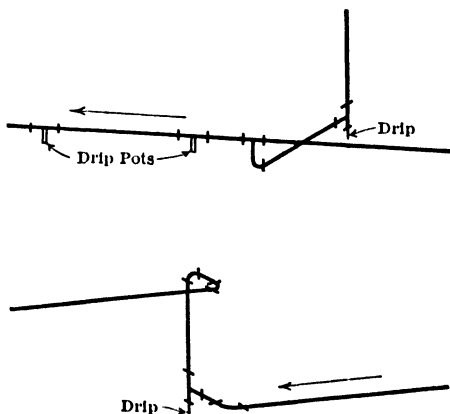


FIG. 116.—Methods of Drainage.

for mains, these may be run horizontally or drained against steam flow by the use of drip pots or short nipples placed on the lower side of the line at intervals, Fig. 116. If branches are taken off from the bottom of the main to a dripped riser they may serve the same purpose.

In mains of any length, as in risers, there must be fixed points at the proper distances on each side of expansion devices so as to fix the way in which expansion occurs and the amount to be cared for. These anchors have the form shown in Fig. 113. These are used on the main to anchor it to the building or structure adjacent to the pipe and not to carry weight like the riser.

The stop valves used on mains and risers are either globe, gate or angle valves, and when possible they should be placed at the high points

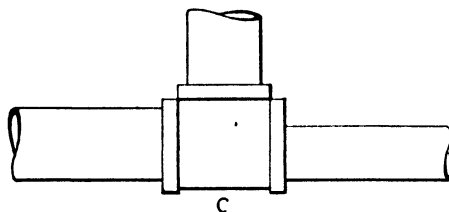
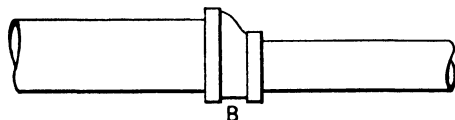
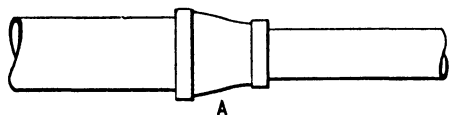


FIG. 117.—Regular and Offset or Eccentric Fittings.

in the line to prevent water of condensation from collecting when closed. If water can collect on one side of a closed valve, the line should be separately dripped at this point through a small petcock to remove the water before opening the valves. Globe valves offer much resistance to flow of fluids, and when used on horizontal lines they should be placed with their spindles in a horizontal plane. This prevents the damming up of condensate by the partition of the valve. With gate valves on such lines the spindles should point upwards.

To prevent the formation of pockets which hold back water in a horizontal main where sizes reduce, eccentric fittings, Fig. 117, are used. The regular reducer *A* is replaced by an eccentric or offset reducer *B*, or an eccentric *T* may be used where a branch line is taken off.

The return main is connected to the boiler in the gravity return system at a point below the water line of the boiler as shown in Fig. 118, through the so-called Hartford connection or underwriter loop. The purpose of this loop is to eliminate the use of check valves in the return line which are necessary to prevent the discharge of all water from the

boiler into the return main whenever pressure is raised in the boiler with the outlet stop valve closed. The pipe *A* is connected to the steam space and water space of the boiler and the return line joins it at a point above the heating surface of the boiler but below the usual water level. In this way the heating surface of the boiler remains covered, as no water below the level of the point of juncture can be driven into the return line from the boiler when there is a great increase of pressure due to a closed stop valve, although at this time steam would enter the return lines.

Without air valves on the radiators, it is necessary to relieve the system of air which is carried in with the supply water or is drawn into

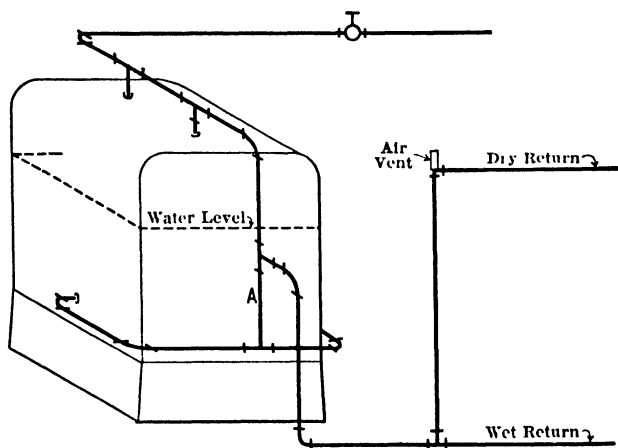


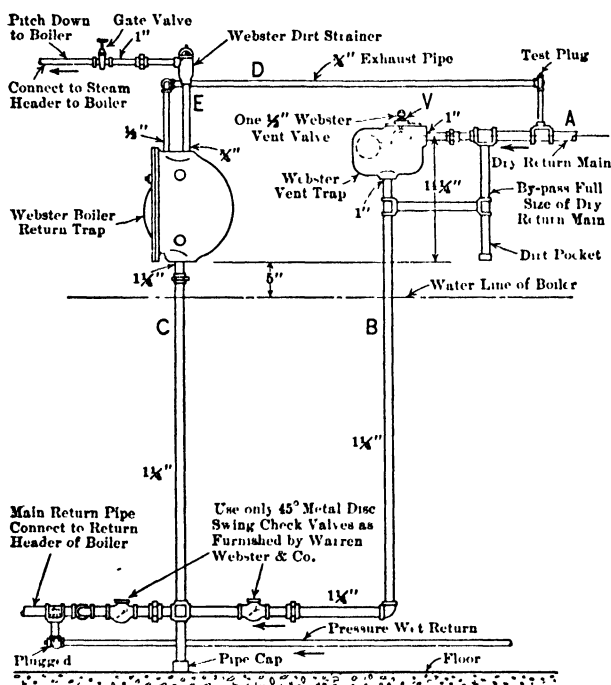
FIG. 118.—Returns with Hartford Connection.

the system when a vacuum exists in the radiators. To relieve this air in a wet return, an air valve is installed at points on the risers, but with the dry return, air valves are installed at a point in the dry return near the boiler, or an automatic device, Fig. 119, may be used.

The return drip from *A* discharges into *B* by the bypass, and if there is sufficient head above the water line in the boiler this opens the two check valves and return water enters the boiler. Air leaves the system by ball valve *V* on the vent trap. If the pressure in the boiler is too great for this return then the water rises in pipe *B* and a float will close the inlet to the ball vent before water can issue. The water also rises in pipe *C* and finally lifts a float in the return trap, closing the exhaust pipe from this trap to the return pipe and opening a valve connected to the steam line *E*. This admits boiler steam which exerts its full pressure on the surface of the water, and the extra head of water forces the return

into the boiler passing the left-hand check valve. The right-hand check valve prevents the passage of water back into the pipe *B*. When the level in the return trap falls, the float cuts off the steam supply and opens a valve to the line *D*.

In several systems the dry return is vented to the atmosphere by carrying one or more open return risers to the roof. In this case the return water must be delivered into the leg of a water seal in which water can rise to a distance above the water line of the boiler such that the



Courtesy of Warren Webster and Co

FIG. 119. — Return Devices.

extra pressure of 0.43% to the foot, due to this rise, is equal to the difference between the boiler pressure and that of the line. If this height is not available, then the device of Fig. 119 is employed. It is well to remember that about 28 in. of water elevation are required to produce a differential of 1 lb per sq in. (about $1\frac{3}{4}$ in. for 1 oz per sq in.).

When a boiler feed pump is used, the return is taken to an air-vented receiver which discharges into the suction of the pump. The receiver must then be placed above the level of the pump as any suction may cause the hot water to vaporize at the reduced pressures. Modern

boiler feed pumps, especially those for independent heating systems of low pressure, are usually of the centrifugal or rotary type which may be operated by electric motors, and in certain cases the low-pressure steam of the heating plants may drive a steam turbine to operate such pumps.

To make circulation more positive and to reduce the pressure at which steam must be supplied to cause proper circulation in the system, vacuum pumps have been applied to the returns. This so-called vacuum system, as installed many years ago by Warren Webster and Company, was intended for plants in which exhaust steam from non-condensing engines was used for the heating of factories, hotels or office buildings. The use of this system permitted the prime movers to exhaust at pressures no greater than those of normal free exhaust, and even lower back pressures might be found on the engines discharging into such systems due to the vacuum. The effectiveness of the vacuum system in giving positive circulation has proved its value and it is used with systems supplied from independent heating boilers.

The Webster Vacuum System of Steam Heating is illustrated by Fig. 120, showing in diagrammatic form a complete system with direct radiators, a convector, an enclosed radiator, a unit heater at one window, a unit heater suspended from ceiling, a hot-water generator heated by a steam coil and even a blast heater tempering the fresh-air supply for ventilation. There is downfeed from an overhead steam main on the left and upfeed risers from a basement main on the right. Radiator branch connections for the radiators on the second floor are concealed by the hung ceiling, and the steam riser on the left and the return risers in the center are concealed. Open risers are shown at the right. On all lines there is ample pitch for drainage. Drainage with the removal of condensate and allowance for expansion are two of the most important items to be considered in laying out steam piping for any purpose. Warren Webster equipment is shown throughout with drip valves from all devices discharging into the main dry return or into the wet return along the floor. The dry return finally discharges into the wet return and the water and air from this are lifted to the receiver tank by the vacuum produced by the motor-driven vacuum pump beneath the tank. This pump delivers the return condensate into the boiler through a line which, although entering at the bottom of the boiler, is also connected above to the steam line from the boiler for balancing purposes.

In the vacuum system water is lifted some distance over obstructions and also from low points if the water can be broken up into slugs. The lift fitting is shown in Fig. 121. Water falls into the low part of this, sealing off the air and preventing its removal from the lower horizontal line. The stoppage of this air flow results in an increase in the air

pressure of the line and water rises in the fitting. When air finally bubbles through this seal the water is broken up into slugs and lifted with ease. When the return line of a vacuum system must be permanently

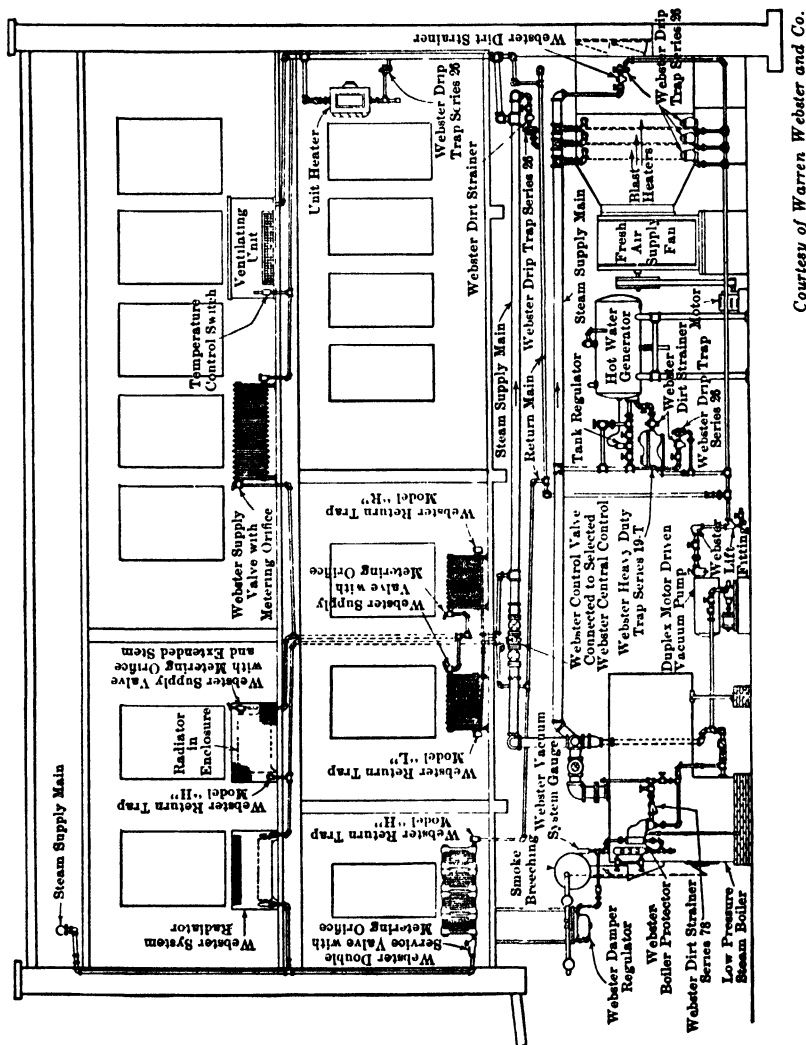


Fig. 120.—Warren Webster Vacuum System of Heating.

depressed at some point to clear obstructions, or openings, it is customary to connect an air pipe from one side of the depression to the other, running this above the obstruction in any way to care for the air flow and allow the drip to find its level at the other side of the depression by gravity. In this way the water and air are delivered. Lift fittings

could be used for this purpose, although this arrangement of pipes is better and offers less resistance.

To protect valves and other devices strainers are placed in the steam or return lines. These, Fig. 122, consist of cast bodies in which metal

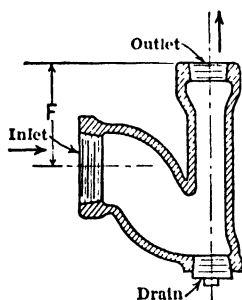


FIG. 121.—Webster Lift Fitting.

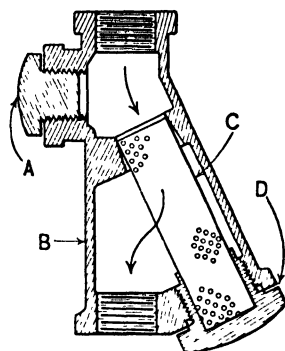
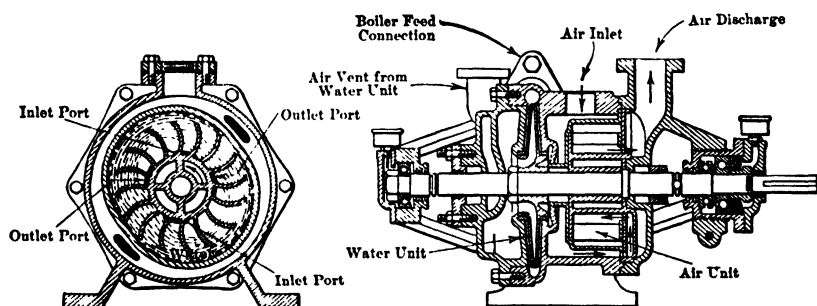


FIG. 122.—Webster Strainer.

basket strainers may be placed and renewed or cleaned by the removal of plug caps.

Another widely used vacuum return system is that equipped with the Jennings vapor vacuum heating pump built by the Nash Engineering Company. The pump, Fig. 123, which draws return water from a



Courtesy of Nash Engineering Co.

FIG. 123.—Jennings Air Pump and Water Pump.

receiver, is made up of two parts: one an air unit which draws air from the top of the receiver and compresses it to atmospheric pressure, the other a water unit which takes condensate from the bottom of the receiver and delivers it into the boiler feed line. The air pump consists of a rotating mass of water driven by vanes, and in the rotation of this body of water, the mass moves outward from the hub of the vanes, suck-

ing in air behind it from a port opposite the side of the vanes at this time, and then as this water is forced inward by the casing, the air below the water ring is compressed and discharged, cross motion of the air being prevented by the vanes, the sides of the casing and the movable piston of water. The water pump is a single-stage centrifugal pump and receives its water at the eye of the impeller from a connection to the bottom of the receiver. Air which may come in with the water is vented back to the receiver. The passages for the air vent from the water unit, the air inlet from the top of the receiver to the air pump, and the air discharge are formed by three divisions of a rectangular cast-

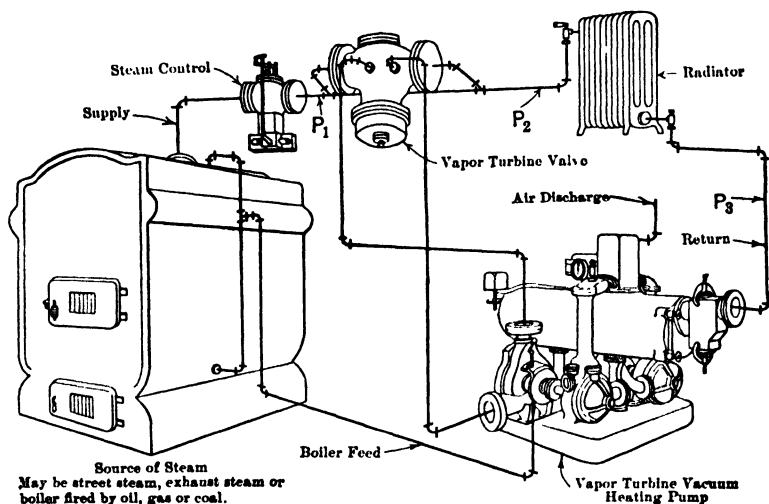
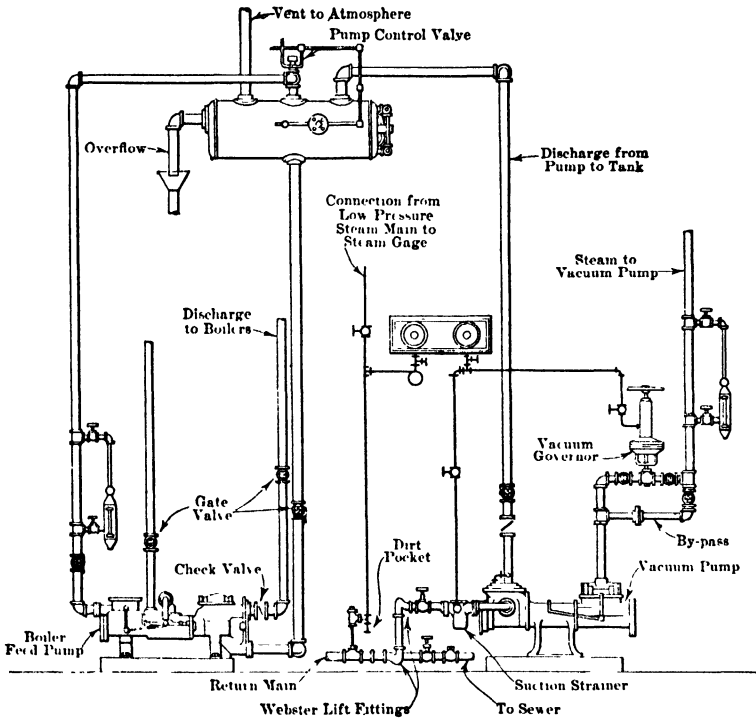


FIG. 124.—Nash Vacuum System.

iron pipe connecting the casing of the pumps, the receiver and the air discharge box. These pumps are driven by an electric motor which is started by a ball float in the receiver at high water or by a vacuum regulator which operates whenever the pressure exceeds the value fixed by the adjustment of the vacuum regulator. The latest improvement in this unit is the operation of the pumps by a steam turbine using the steam as it is delivered into the heating system. This requires a pressure difference of $2\frac{1}{2}\%$ on the turbine which is maintained by a special valve.

The arrangement of these parts is shown in Fig. 124. Steam from the boiler passes the vapor turbine valve, which produces a continuous drop in pressure of $2\frac{1}{2}\%$ regardless of the pressure in the return. Steam from the high side of this is taken to the turbine inlet at the top of the

unit and drives this continuously, the exhaust being returned to the low side of the turbine valve for use in the system. The return condensate from the system is brought into the receiver through a large strainer at the right and flows to the pump at the base, being delivered into the boiler feed line as shown. The compressed air is led to the top of the unit for discharge into an exhaust line. In the unit shown two pumps are connected to the receiver. The pump back of the receiver is a spare



Courtesy of Warren Webster and Co.

FIG. 125.—Return from Vacuum Pump to Receiver for Boiler Feed Pump.

unit and is driven by an electric motor which is automatically controlled. The turbine-driven pump operates continuously, and should it fail to remove condensate fast enough, a float inside of the receiver rises and operates a switch at the left end of the receiver to start the motor-driven unit. Should the vacuum fall below the desired point, although water is being removed, then a vacuum regulator on top of the receiver at the center starts the motor of the rear unit. The water pumps will deliver water against 20%, and if higher boiler pressures are used in a plant this unit discharges into the boiler feed receiver of the plant. Such an

arrangement for a Warren Webster direct-acting vacuum pump is shown in Fig. 125, the receiver being mounted at a high level.

Pipe covering is used on pipes placed in chases or run in the open not only to conserve heat but also to make the spaces through which they pass of livable temperatures. Coverings are usually made of materials containing many small air cells in molded forms 3 ft in length and split for application to a pipe, Fig. 126. The sections are covered with canvas which is pasted around the joints and then thin metal bands are applied. The sectional coverings are made of different thicknesses, 1 in. being the approximate thickness of standard covering, although for large pipes thicknesses of $1\frac{1}{2}$ in. or more may be used. The materials used are 85% magnesia (hydrated basic carbonate of magnesia with asbestos fibre), asbestos mixtures, hair felt, diatomaceous earth, as

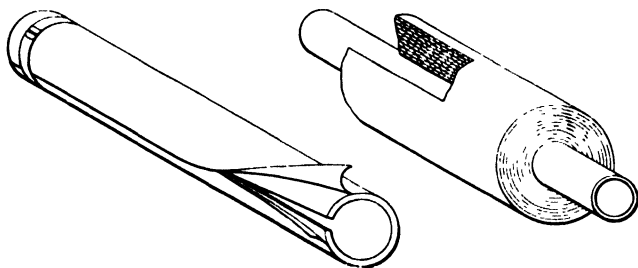


FIG. 126.—Pipe Covering Sections.

well as wool felt for hot and cold water lines. As shown in the figure, a cheap form of covering is one in which corrugated asbestos sheets are placed in alternate layers in order to include larger air cells, forming a lighter and less expensive covering. This is known as air-cell covering. The latest form of pipe insulation is made from several thin sheets of crumpled aluminum foil laid around the pipe. These are placed beneath a bright metal jacket held concentric with the pipe by distance pieces called towers. Three sheets of crumpled foil are used to the inch of thickness.

For flat surfaces and very large pipes, insulating blocks, 6 in. \times 36 in. or 3 in. \times 18 in. and of varying thickness, are wired in position, after which a finish of hard plaster is applied. Eighty-five per cent magnesia plastic is made of the same material as the covering and is applied to irregular surfaces or between blocks as a mortar or plaster after mixing with the proper amount of water.

Pipe is covered not only to prevent heat loss but also to prevent the addition of heat to lines carrying cold refrigerant in air-conditioning

apparatus and to prevent the sweating of these pipes (condensing moisture from air) as they lead to the conditioner. Coverings for such pipes are made of felt or cork.

Tests have been conducted by many investigators to determine the value of U for pipe covering in the heat-transmission formula

$$q = UA(t_s - t_r) \quad (6-6)$$

in which A is the external area of the pipe, t_s the temperature of the substance in the pipe and t_r the temperature of the room. These experi-

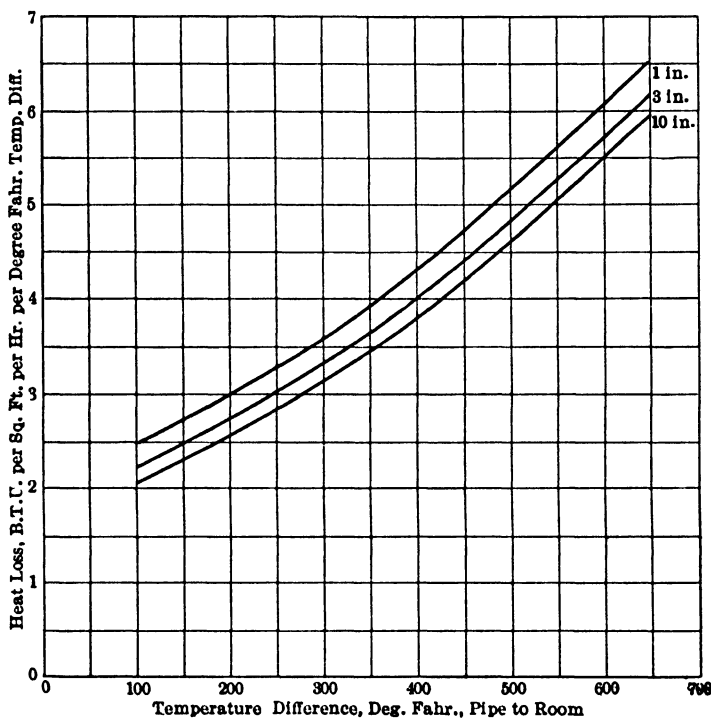


FIG. 127.—Curves of Heat Loss from Bare Pipes from Tests by Heilman.

ments have shown that U depends not only on the thickness of the covering but also on the difference between the temperature of the substance and that of the room. Even the value of U for a bare pipe, Fig. 127, varies with temperature difference and size as shown by the experiments of Heilman.* The reason for the variations in the value of U for the covering with size depends upon the fact that heat is being transmitted

* Heat Loss from Bare Pipes at High Temperature, by A. L. Heilman, Transactions A.S.M.E., Vol. 44, p. 299.

through an increasing area as it passes through the circular covering. From the principles of Chapter III,

$$q_t = U 2\pi \frac{r_1}{12} l(t_s - t_r) = - \frac{k}{dr} 2\pi \frac{r}{12} l dt \quad (6-7)$$

but

$$q_t = q 2\pi \frac{r_2}{12} l$$

r_1 = radius of pipe, inches;

t_r = temperature of room, degrees F;

t_s = temperature of steam, degrees F;

l = length, feet;

q = the heat delivered per square foot of canvas covering per hour at radius r_2 .

Hence,

$$2\pi q \frac{r_2}{12} l = - \frac{k}{dr} 2\pi \frac{r}{12} l dt$$

$$qr_2 \frac{dr}{r} = - k dt$$

Integrating this between r_1 and r_2 for which the temperatures are respectively t_s and t_2 , there results

$$qr_2 \log_e \frac{r_2}{r_1} = k(t_s - t_2) \quad (6-8)$$

This gives

$$q = \frac{k(t_s - t_2)}{r_2 \log_e \frac{r_2}{r_1}} \quad (6-9)$$

and

$$U = \frac{qr_2}{r_1(t_s - t_r)} = \frac{k(t_s - t_2)}{r_1(t_s - t_r) \log_e \frac{r_2}{r_1}} \quad (6-10)$$

Now t_2 differs from t_r by an amount d which has been determined by Heilman for different sizes of covering and different values of heat flow per square foot, from which results he derived the general empirical equation

$$d = \frac{272.5q}{q + \frac{564}{D^{0.19}}} = \frac{272.5}{1 + \frac{564}{qD^{0.19}}} \quad (6-11)$$

$d = (t_2 - t_r)$ in degrees F;
 q = Btu per hour per square foot of canvas surface;
 D = outer diameter of canvas surface, inches.

Heilman* has also determined the value of k for different types of

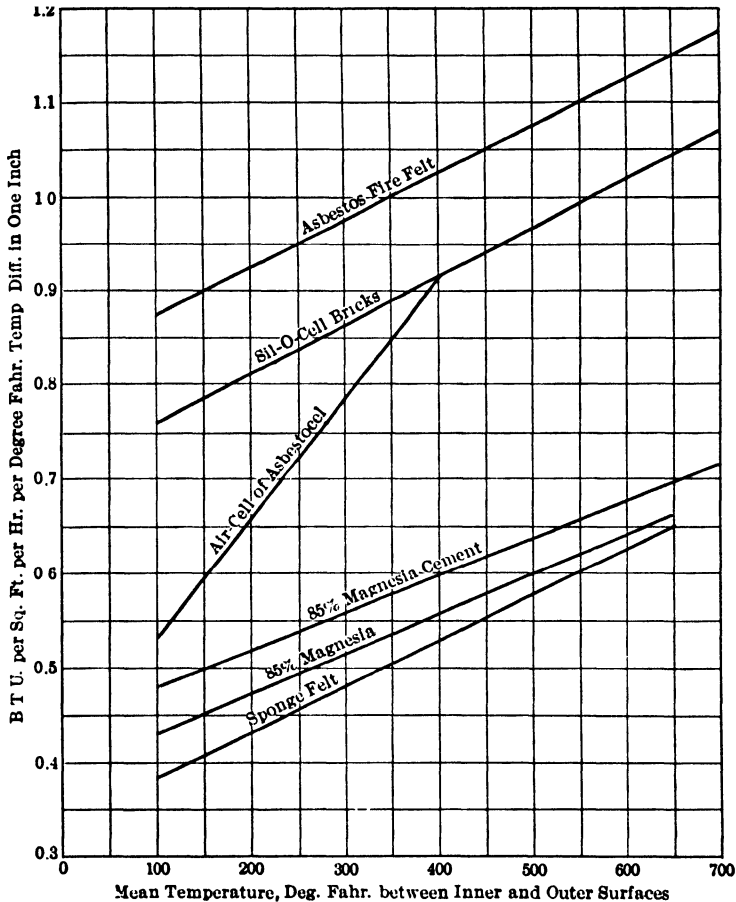


FIG. 128.—Coefficients of Conduction of Heat Insulating Materials from Tests by Heilman.

covering at various mean temperatures, $\frac{t_1 + t_2}{2}$, so that the heat per foot of length can be determined as

* Heat Losses through Insulating Materials, R. H. Heilman, Mechanical Engineering, October, 1924, p. 593.

$$\begin{aligned}
 q_t &= U \frac{2\pi}{12} r_1 (t_s - t_r) \\
 &= 2\pi \frac{k(t_s - t_r - d)}{12 \log_e \frac{r_2}{r_1}} \quad (6-12)
 \end{aligned}$$

The values of k averaged from Heilman's work are given at various mean temperatures in Fig. 128.

From this the following table of values for heat loss per foot per hour has been worked out for steam at 215° F and at 5# gage pressure and for water at 180° F with room temperature at 70° and 40°. In addition, the heat per square foot of flat covering has been worked out determining d by assuming a 40-in. radius in the formula for this quantity.

TABLE 6-13

HEAT LOSS PER LINEAL FOOT OF PIPE COVERED WITH 1 IN.
STANDARD THICKNESS OF COVERING

85% Magnesia and Air Cell. Btu per hr

Nominal Pipe Size	Temperature Range, Degrees F											
	215°-70°		215°-40°		227°-70°		227°-40°		180°-70°		180°-40°	
	85%	A.C.	85%	A.C.	85%	A.C.	85%	A.C.	85%	A.C.	85%	A.C.
$\frac{3}{4}$	28	36	33	42	30	40	36	45	21	27	27	34
1	32	40	37	47	34	46	40	51	24	29	30	38
$1\frac{1}{4}$	37	47	43	55	39	53	47	59	28	33	34	43
$1\frac{1}{2}$	39	50	47	58	42	56	51	64	30	37	37	47
2	46	58	55	69	50	65	59	75	35	42	45	54
$2\frac{1}{2}$	52	66	63	78	57	74	68	86	40	48	50	62
3	61	77	72	91	65	86	78	98	46	55	57	71
4	74	92	88	110	79	103	95	120	56	57	70	86
5	88	111	107	132	95	123	113	143	67	80	83	102
6	102	127	123	152	108	142	132	165	77	93	96	118
8	128	158	155	192	137	177	165	208	97	117	120	147
10	156	193	189	234	167	217	202	255	118	142	147	178
12	182	226	222	273	195	253	236	297	137	164	170	208
14	198	246	242	298	213	276	257	323	150	179	186	227
16	224	278	273	338	241	313	290	365	169	203	210	256
Flat per sq ft	47	58	56	70	51	68	61	75	34	42	43	53

The values of U found for some thicknesses may be used to determine U' for other thicknesses or temperature ranges as follows:

$$\frac{U'}{U} = \frac{(t'_s - t'_2)(t_s - t_r)r_1 \log_e \frac{r_2}{r_1}}{(t_s - t_2)(t'_s - t'_r)r'_1 \log_e \frac{r'_2}{r'_1}}$$

$$t_2 = t_r - d$$

$$t'_2 = t'_r - d'$$

The values of d' and d must be determined by a series of approximations.

If covering of radii r_3 and r_2 and conductivity k_0 is used over another covering of radii r_2 and r_1 and conductivity k_1 the heats transmitted per square foot of outer surfaces are related by

$$q_0 = \frac{q_1 r_2}{r_3} = \frac{k_0(t'_1 - t_2)}{r_3 \log_e \frac{r_3}{r_2}} = \frac{k_1(t_s - t'_1)}{r_3 \log_e \frac{r_2}{r_1}}$$

and

$$q_0 r_3 \left[\frac{1}{k} \log_e \frac{r_2}{r_1} + \frac{1}{k_0} \log_e \frac{r_3}{r_2} \right] = t_s - t_2$$

Now

$$q_0 = U \frac{r_1}{r_3} (t_s - t_r)$$

hence

$$U r_1 (t_s - t_r) = q_0 r_3$$

or

$$U = \frac{(t_s - t_2)}{\left(\frac{1}{k_1} \log_e \frac{r_2}{r_1} + \frac{1}{k_0} \log_e \frac{r_3}{r_2} \right) r_1 (t_s - t_r)} = \frac{(\text{mean } k)(t_s - t_2)}{r_1 (t_s - t_r) \log_e \frac{r_3}{r_1}}$$

$$\text{Mean } k = \frac{\log_e \frac{r_2}{r_1} + \log_e \frac{r_3}{r_2}}{\frac{1}{k_1} \log_e \frac{r_2}{r_1} + \frac{1}{k_0} \log_e \frac{r_3}{r_2}} \quad (6-14)$$

For flat plate the formula for q_t is

$$q_t = \frac{k}{L} A(t_s - t_2) = UA(t_s - t_r)$$

Heilman suggests that the values of d be computed for a diameter of 30 in. in getting the U for flat plates. The author has used a diameter of 80 in. in finding d for the table.

The most economical thickness of covering for a given condition of temperature is determined by computing the sum of the cost of steam condensed per season by the heat loss per foot of length with different thicknesses and the yearly fixed charges of the cost of these and using that thickness which gives the minimum total season cost. The conditions vary with different installations, and as a guide it may be stated that standard 1-in. thickness of 85% magnesia should be used on pipes carrying fluids below 150° F, while 2-in. thickness should be used with fluids at 280° F. These coverings reduce the heat loss from bare pipes by 80% to 90%.*

Initial Heating and Effect of Uncovered Pipes. Although it has been shown by experiment that a radiator will condense almost 1 lb of steam per hr per sq ft for a very short time when steam is first turned into a radiator, no allowance is made for this in the design of normal piping as such action will occur only on a few radiators simultaneously. It is noted also that, although uncovered pipes add to the radiating surface, if there has been no reduction in radiation surface for the heat demand due to these pipes their presence reduces the steam needed by the radiators and does not increase the steam supply for the building.

Equivalent Temperature of Building. At times it may be desired to check the design of a heating system by a test to ascertain its performance at some outside temperature differing from that used in the calculations as such an actual outside temperature cannot be obtained. Let the computed heat loss from the room to be tested be called q_{RD} for a room temperature t_{RD} and an outside temperature t_{OD} , while the actual outside temperature at the time of the test is t_{OT} . It will also be assumed that the temperature of the steam is t_{ST} in place of the t_{SD} which was used in the design. If the heat per hour delivered per square foot of radiators of total area A_D in the design were q_D the heat per hour per square foot for other temperature conditions may be found from the relation

$$q_T = q_D \left(\frac{\Delta t_T}{\Delta t_D} \right)^{1.3} \quad (6-16)$$

The room temperature which should be found on the test, t_{RE} , as equivalent to t_{RD} , is given by the heat equation

* See page 666 of A.S.H.V.E. Guide for 1936 for diagram by which the economical thickness may be determined. Also, *Advantages of Heat Insulation* by R. H. Heilman, *Trans. A.S.H.V.E.*, Vol. 32, 1926.

$$q_{RD} \frac{(t_{RE} - t_{OT})}{(t_{RD} - t_{OD})} = A_D q_D \left(\frac{t_{ST} - t_{RE}}{t_{SD} - t_{RD}} \right)^{1.3}$$

$$\frac{(t_{SD} - t_{RD})^{1.3}}{(t_{RD} - t_{OD})} (t_{RE} - t_{OT}) = (t_{ST} - t_{RE})^{1.3} \quad (6-17)$$

$$= (\text{Const})(t_{RE} - t_{OT}) = (t_{ST} - t_{RE})^{1.3} \quad (6.18)$$

This equation may be solved by successive approximations or by plotting the two sides of the equation against t_{RE} . The point of intersection will give the desired point.

For a room of 24 Mbh at 70° F in zero weather with 100 sq ft of E.D.R. with 215° F steam, the equivalent temperature with 30° F outside would be

$$\frac{(215 - 70)^{1.3}}{70} (t_{RE} - 30) = (215 - t_{RE})^{1.3}$$

$$9 \ 24(t_{RE} - 30) = (215 - t_{RE})^{1.3}$$

Try 80 $9 \ 24 \times 50 = 462 \neq 600$

Try 89 $9 \ 24 \times 59 = 545 = 545$

This means that with complete heating of the radiators in this room in 30 degree weather the room should reach 89° F.

Each room should be calculated separately, but as a guide the following table has been worked out for certain rooms using steam at 5# pressure and a design based on zero degree weather.

TABLE 6-14
EQUIVALENT INSIDE TEMPERATURE

Actual Outside Temperature	Designed Inside Temperature, Degrees F, with 0° F Outside			
	50°	60°	70°	80°
-10° F	43	53	63	74
0	50	60	70	80
10	57	67	76	85
20	65	74	83	92
30	72	81	89	98

PROBLEMS

1. A riser of an upfeed single-pipe system has radiators on four floors of the following amounts of E.D.R.: 1st floor, 60 sq ft and 30 sq ft; 2nd floor, 100 sq ft and 40 sq ft; 3rd floor, 25 sq ft; 4th floor, 80 sq ft and 40 sq ft. Make oblique drawing of riser and radiators, and indicate sizes and forms of radiator runouts, riser sizes and riser runout. Assume any desired drop in pressure.

2. Make a riser diagram for same radiators for a downfeed double-pipe system. Assume drop in pressure.

3. A steam main supplies the various risers, starting at *A*, on which the following amounts of E.D.R. are found: *A*—300 sq ft; *B*—120 sq ft; *C*—200 sq ft; *D*—700 sq ft; *E*—800 sq ft; *F*—375 sq ft. Give size of main at each section. Give size of return main which starts at *A* and connects to boiler beyond *F*. Assume drop in pressure. Why is it unnecessary to give length in this problem and in problems 1 and 2?

4. Data in problem 2 are for the riser *F* of problem 3 and the floors are 13 ft apart, while risers are 20 ft apart, and riser *A* is 25 ft from boiler with 5 elbows. Make oblique sketch of supply and return main and riser *F* and find the equivalent length from boiler to fourth floor radiator on *F*. Also find equivalent length of return from *F* to boiler using 4 ells in this line. What is the total drop of pressure in this circuit?

5. Select direct radiators for heat demand for room of problem in Chapter V.

6. A building of 20,000 sq ft of E.D.R. steam radiation is to be supplied by steam at a mean pressure of 40% gage. The line is 500 ft long and contains 6 elbows. The drop permitted in this line is 2%. Find size of main to carry this steam by using Table 6-5 after finding the equivalent surface if the drop were $\frac{1}{4}$ lb per 100 ft with steam at 215° F. How large would you make the wet return from this building with a length of 500 ft which contains 4 elbows?

7. Find the heat loss per ft of length from a standard 8 in. pipe carrying steam at 80% gage pressure with 85% magnesia covering 2 in. thick. $t_r = 40^\circ$ F.

8. What heat loss is found from pipes in a basement carrying 150 ft of 6 in. mains at 215° F with 85% magnesia covering? Find the loss as a percentage of the heat in the steam carried by this main with 0.15% drop per 100 ft. Thickness 1 in.

9. A room was designed to operate at 72° F in 10° weather. What temperature should be found in this room at 20° outside temperature? Steam at 1% gage.

CHAPTER VII

HOT WATER HEATING

In hot water heating the liquid is cooled from 10 to 30 or more degrees in passing through the radiator (the mean water condition being 170° F in a room at 70° F for standard rating). To transmit 150 Btu per hr per sq ft, the standard transmission from 1 sq ft of hot-water radiation, it is seen that approximately 1 gal of water must be supplied per sq ft per hr if there is a 20° F drop in the water passing through the radiator. Although conditions will alter this number, 1 gal per hr per sq ft of hot-water surface in the hot-water system is an average number to remember for the total hourly rate of circulation in the system. This is similar to $\frac{1}{4}$ lb of steam per sq ft of direct radiation per hr in steam systems.

Twelve pounds of water cooling through 20° F would be required for the heat equal to that from the condensation of $\frac{1}{4}$ lb of steam, and this water at 180° F would occupy 0.20 cu ft, while $\frac{1}{4}$ lb of steam at 215° F would occupy 6.34 cu ft, or 31.7 times as much as the volume of water. From this it is seen that for a given amount of heat supply the steam must travel very much faster than the water or the water may be carried in smaller pipes. In hot-water installations with forced circulation the mains and risers will be quite small for a given capacity when compared with those for low-pressure steam. It must be realized also that if the water is put under pressure so that within the radiator the mean temperature is 215° F, just as much heat per square foot per hour may be obtained from the radiator with water as with steam.

By the use of lower temperatures than those found in steam radiators the heating by hot water is less intense in the vicinity of the radiators. Moreover, in mild weather, the temperature of the hot-water supply may be low so that the whole radiator at this time is slightly above room temperature. In the steam system the top part of the radiator is always heated much above room temperature even in mild weather. In both systems the amount of heat from the radiator is the same, but with steam the small heated area at a high temperature forms an intense center for the radiation of heat, whereas the whole radiator with water is at a temperature slightly above that of the room.

The radiators arranged with supply risers and return risers attached to the same flow main, as shown in Fig. 129, form the single-pipe or one-pipe system. In this case the portion of the hot water taken by the first set of radiators is returned to the flow line after giving up heat to the radiators. On account of the reduced temperature at outflow of this water, the mixture of this with the other water of the flow main gives a lower temperature of supply for the second set of radiators. The temperature in the flow main is gradually decreasing and hence for equal amounts of heat, radiators in the further sections of the heating sys-

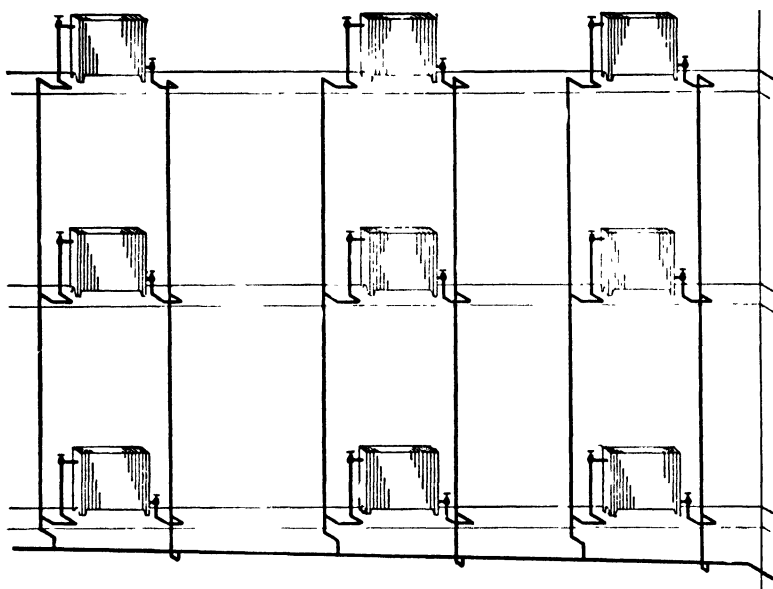


FIG. 129.—Single Pipe or Single Main Hot-water System of Heating.

tem must be larger than those of the first set nearest the hot-water heater. In some single-pipe systems the full water circulation is sent through each radiator in succession. This would give the same temperature on the final radiator as that described above.

The supply riser may be taken from a point near the top of the flow line in the one-pipe system by the use of an eccentric tee or a Y and may be brought back to the lower part of the line by using similar fittings.

The hot-water type of radiators (sections connected at top and bottom hubs) are equipped with valves arranged to give full opening by a quarter or half turn of the handle, and key-operated air cocks at the top of the radiator remove collected air when necessary.

To give the same temperature of supply for each set of risers, two

flow pipes are used, the supply flow pipe and the return flow pipe. By arranging these risers and flow pipes, Fig. 130, with the supply riser nearest the boiler connected with the radiators which discharge into a return riser connected to the end of the return flow line most remote from the boiler, the total length of circuit to each set of radiators is approximately the same. This is known as the reversed return arrangement of the two-pipe system. In the direct-return arrangement both risers of a given set of radiators are connected at practically the same distance from the heater, giving various lengths of circuits for the differ-

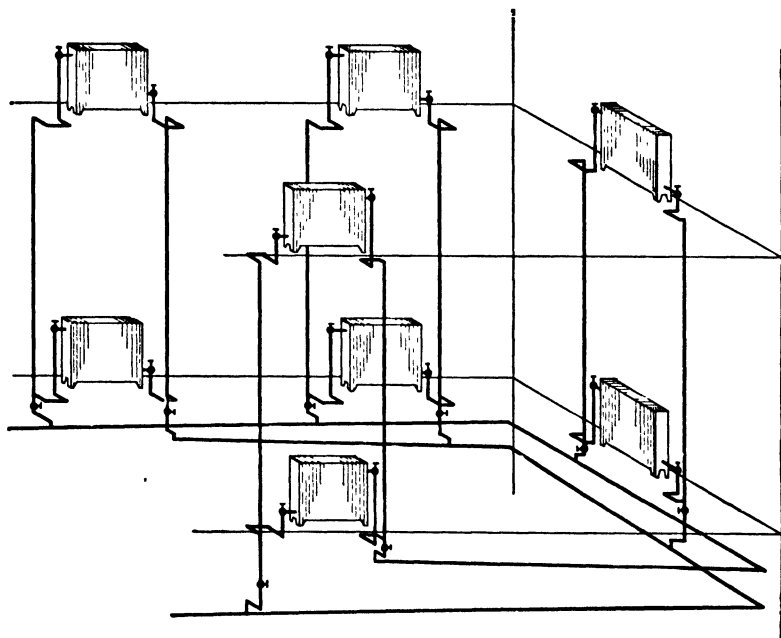


FIG. 130.—Reversed Return Two-pipe Hot-water System of Heating.

ent risers, thus tending to unbalance the flow in the various stacks of radiators. Those near the boiler with a shorter circuit than any other set have less resistance to the flow of a given quantity of water. The piping of the Mills system for steam could be used for hot water by removing the drips at the ends of the downfeed risers and connecting the return risers to a return flow main arranged for reversed return.

A number of changes may be made in the piping arrangement, but the single-pipe system and the two-pipe system with reversed return are the most common.

The greatest drawback to the hot-water system is the difficulty of

equalizing the flow at all times. Experiments have been made which show that in systems which were carefully worked out for balance at one load the radiators on different risers and even on the same riser heat up at widely varying rates, even reverse flow occurring in certain radiators.

Runouts are taken from the flow mains as shown in Fig. 131 when there is sufficient length in them to permit the small expansions which occur in this system. If the connection is a short one, the swiveling

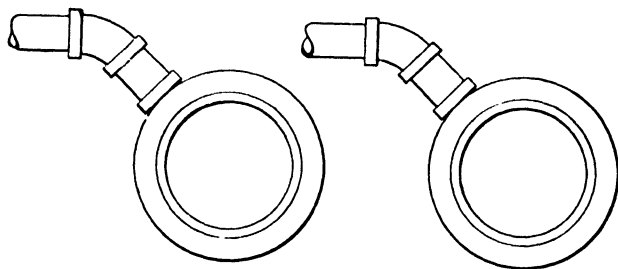


FIG. 131.—Branch or Runout Connections.

elbow arrangement, Fig. 132, must be used. The former arrangement has much less friction than the latter and moreover it requires less head room for its installation, both important items in the gravity systems of hot-water heating.

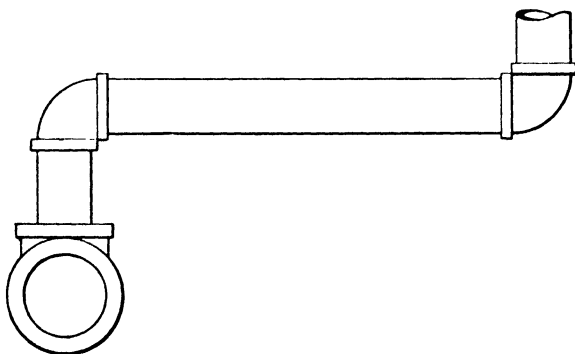


FIG. 132.—Branch or Runout Connection.

With branch connections for risers or radiators care must be taken so to proportion the parts that there is no tendency for water to pass into one branch or pass beyond it and so disturb the flow. Orifices or extra elbows are used in risers to throw extra resistance into that portion which might short-circuit a lower radiator connection or other part.

The insertion of an orifice plate, Fig. 133, in the line tends to prevent by-passing, the opening being so selected that it increases the resistance in the line to upper radiators and uses part of the head set up by the height of the riser pipes.

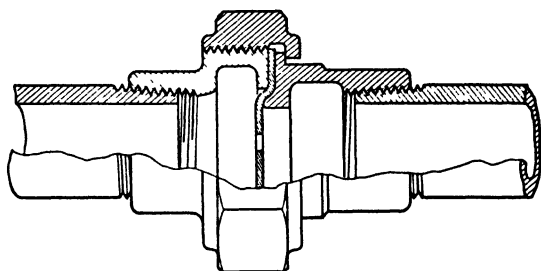


FIG 133.—Hot Water Orifice Restriction.

The drop in pressure across an orifice is determined from the loss due to the sudden contraction and sudden enlargement of the stream occurring at the orifice. Theoretical considerations show that this loss is closely given by an equation

$$h = c \frac{V^2}{2g} \quad (7-1)$$

From experiments by Giesecke on orifices for hot-water systems reported in Trans. A.S.H.V.E. for 1930 the loss at an orifice depends on the ratio of area of opening to that of pipe and the velocity, and the author has reduced the values of c' , to be used in an average formula, for different ratios of area of orifice to area of pipe, A_o/A_p .

$$h \text{ in mi} = c' V^2; \quad (7-2)$$

V = velocity in pipe in inches per second;

mi = milli-inches* = $\frac{1}{1000}$ inch, abbreviated as mi.

TABLE 7-1
ORIFICE COEFFICIENTS

Ratio A_o/A_p .	0.082	0 1	0.15	0 20	0 25	0.30	0 35	0 40	0.45	0.50	0 55	0.60
c'	422	280	110	51	28	16	10	6.2	4.0	2.7	2.0	1.7

If the milli-inches to be absorbed in orifice friction are known, as well as the velocity, then the value of c' can be computed and the value of A_o/A_p may be taken from the curves of Fig. 134. The diameter of the orifice, d_o , may then be found from the actual inside diameter of pipe, d_p .

$$d_o = d_p \sqrt{A_o/A_p} \quad (7-3)$$

* This is called milinches in the A.S.H.V.E. Guide.

Water completely fills the hot-water system when operating properly and as the temperature of the water varies throughout the heating season or even in one day as the demands on the system change, the volume of the water will change as the result of its expansion and contraction

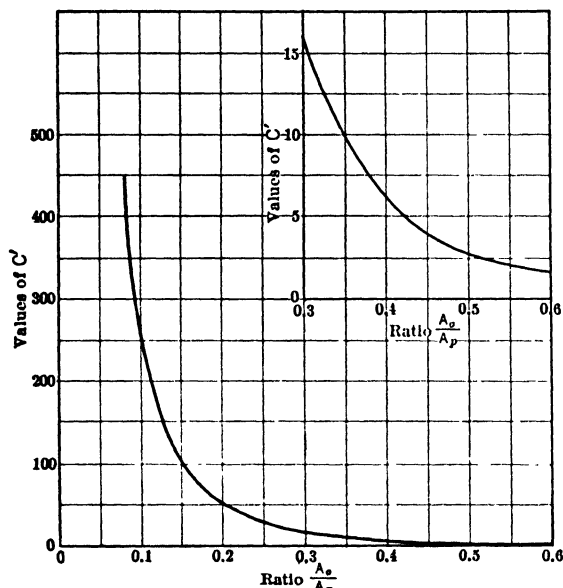


FIG. 134.—Orifice Coefficient.

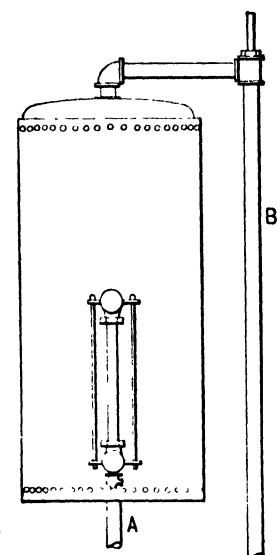


FIG. 135.—Expansion Tank.

with these temperature variations. To allow for this change, expansion tanks, Fig. 135, are placed at some part of the system. The volume of these tanks is fixed by computing the volume change of the water within the system from the probable temperature range.

TABLE 7-2
VOLUME OF 1 LB AND WEIGHT OF 1 CU FT OF WATER
(From Goodenough)

t	Volume 1 lb in cu ft	Weight 1 cu ft in lb	t	Volume 1 lb in cu ft	Weight 1 cu ft in lb	t	Volume 1 lb in cu ft	Weight 1 cu ft in lb
40	0.01602	62.43	140	0.01629	61.38	240	0.01692	59.11
60	0.01603	62.37	160	0.01639	61.00	260	0.01708	58.55
80	0.01607	62.22	180	0.01651	60.58	280	0.01725	57.96
100	0.01613	62.00	200	0.01663	60.12	300	0.01745	57.32
120	0.01620	61.71	220	0.01677	59.63	320	0.01766	56.62

$$\text{Minimum volume of tank} = (\text{pounds of water in system})(V_{\text{hot}} - V_{\text{cold}}) \quad (7-4)$$

V = volume of 1 lb in cu ft

Make tank at least 75% greater than the minimum volume.

This requires the computation of the weight of water to fill the system, including heater, pipes, radiators and fittings, which amounts to about 0.03 cu ft per sq ft of radiator surface in the system. The table below may be used as a guide.

TABLE 7-3
VOLUMES OF EXPANSION TANKS

Size	Gallons Capacity	Square Feet of Radiation	Size	Gallons Capacity	Square Feet of Radiation
10 × 20	7	250	16 × 36	32	1300
12 × 20	10	300	16 × 48	42	2000
12 × 30	15	500	18 × 60	66	3000
14 × 30	20	700	20 × 60	82	5000
16 × 30	26	950	22 × 60	100	6000

The tank is connected with the system by pipe *A* and is placed higher than the highest radiator of the system if the tank is open to the atmosphere. The tank is provided with a vented overflow *B*. It must be located in a space in which the temperature is above that of freezing. In some cases a pipe is taken from the side of the tank and connected with a return riser to permit circulation in the tank and its piping so as to maintain a proper temperature. If the system is a closed one the tank may be located at any level and air above the water in the tank is held at a proper pressure.

It is to be remembered that water boils at 212° F under atmospheric pressure, so that a vented air tank will give off steam when water at this temperature reaches it in its circulation, although the static head in the system below this level has prevented boiling until the tank is reached. The tank must be under a pressure greater than that of boiling water at the temperature within the tank.

The circulation in a gravity hot-water system is due to the difference in weight between the columns of cold water in the return pipes and that of an equal height of the warm water in the supply pipes, while in the

forced or mechanical circulation system, Fig. 136, a small pump produces a pressure difference in addition to that from the hot and cold columns. This small impeller pump requires little energy but it supplies sufficient head to speed the water so that much smaller pipes may be used for flow lines, runouts and risers. The gravity circulating head always exists in hot-water systems. Except for the pressure difference due to the densities of the hot and cold water, the water columns are balanced so that the pump for forced circulation merely supplies extra energy to overcome the friction of the system.

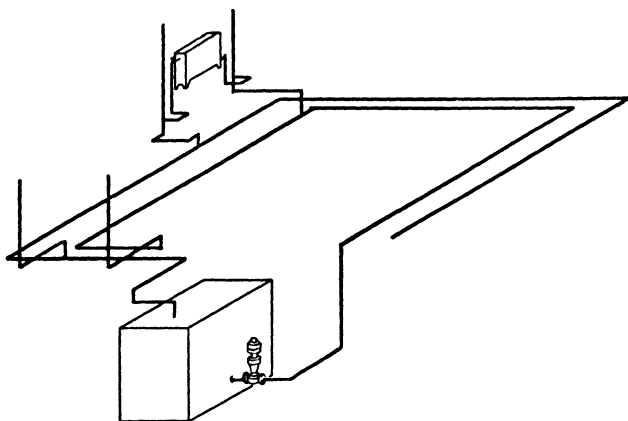


FIG. 136.—Hot-water System with Circulating Pump.

Radiator Sizes. In determining the radiator sizes, three temperatures must be fixed: the temperature of the water entering the particular radiator, that of the water leaving and that of the space to be heated. From the mean temperature of the water in the radiator and that of the room, the Table A-9, page 404, will give the radiator temperature factor by which the standard transmission of 240 Btu per hr per sq ft must be multiplied to give the heat transmission under the particular temperature conditions.

$$\text{Btu per sq ft per hr} = 240 \times \text{Radiator temperature factor} \quad (7-5)$$

If the heat is expressed in the unit of 1000 Btu per hr (abbreviated Mbh), the following is given:

$$\text{Mbh per sq ft per hr} = 0.240 \times \text{Radiator temperature factor} \quad (7-6)$$

The surface of the radiators for a given room would then be

$$\text{Total surface for room} = \frac{\text{Heat requirements in Mbh}}{0.240 \times \text{Temperature factor}} \quad (7-7)$$

If data sheets relating to special forms of radiators or unit heaters give the equivalent square feet of standard hot-water radiation under various conditions of temperatures and of installation dimension, then the method of determining size would be

$$\text{Total hot-water surface per room} = \frac{1000 \text{ Heat requirements in Mbh}}{150 (\text{Temperature factor})} \quad (7-8)$$

If special forms of surface are rated in Mbh per square foot per hour (q_i) under certain conditions then the surface required would be

$$\text{Total special surface per room} = \frac{\text{Heat requirement in Mbh}}{q_i (\text{Temperature factor})} \quad (7-9)$$

This means that, in the design of the system, the temperature variation in the water from boiler to each radiator and the temperature of the return must be known as well as the room temperature, and as this is connected with the amount of heat needed in each room, the radiator size can be determined only after the pipe sizes and water temperatures are calculated. The heat demand of the radiator in any room, however, in Mbh, remains constant in all the computations.

Radiators for Building, Chap. V. With water entering at 180° F and leaving at 160° F the water radiators with a reversed return system for the building of Chapter V would be found from the heat demand for each room or they may be determined from the steam radiator sizes as follows:

From Table A-9 the radiator temperature factor for 180° – 160° – 70° F is 0.617.

$$\text{Heat per sq ft per hr} = 240 \times 0.617 = 148 \text{ Btu per hr}$$

The surface of the radiators for hot water for this building will be found by multiplying the previous steam areas by $\frac{240}{148}$ or 1.62 as the heat delivered will be the same for both systems. If the steam system had not been computed the surface in each room would be computed as follows:

$$\text{Required hot water surface} = \frac{\text{Mbh for room}}{0.148}$$

TABLE 7-4

HOT WATER SURFACE FOR RESIDENCE OF L. Q. SMITH

Room	Radiator	Radiator Surface	Mbh
101	101a	68	10.08
	101b	68	10.08
102	102a	51	7.56
	102b	51	7.56
103	103a	15	2.16
104	104a	44	6.48
105	105a	34	5.04
	105b	34	5.04
106	106a	83	12.23
201	201a	54	7.93
	201b	54	7.93
202	202a	73	10.80
203	203a	24	3.60
204	204a	28	4.20
	204b	28	4.20
205	205a	29	4.32
	205b	29	4.32
206	206a	19	2.64
Total	786	116.17

Pipe Sizes. To determine pipe sizes for hot water systems a riser layout, Fig. 137, is made on which the heat demand for each radiator is marked on the plan. The temperature of the water from the heater

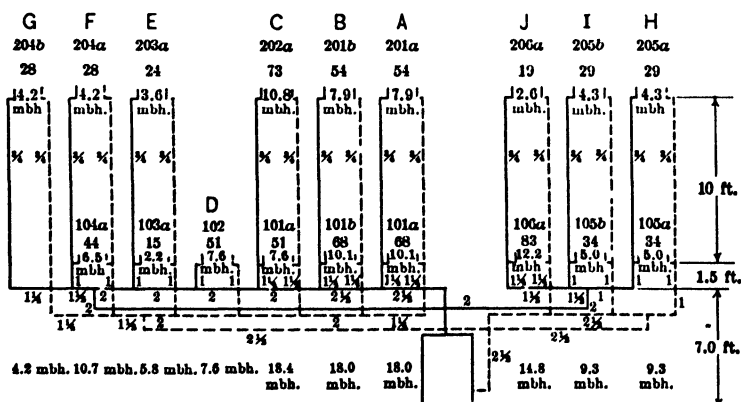


FIG. 137.—Riser and Main Diagram Hot-water System.

is then fixed, and this with proper pipe covering will be approximately the temperature of the water in the flow line at all risers. The temperature of the return water at the heater is then assumed using a drop of 10 or more degrees. The amount of water flowing per hour through the flow line from the heater will be the sum of the heat demands of the complete system divided by the difference in enthalpies of the liquid at the temperatures at the heater outlet and return point; the quantity of water is given as

$$W \text{ lb per hr} = \frac{1000 (\text{Heat requirement in Mbh})}{h_{fs} - h_{fr}} \quad (7-10)$$

$$\begin{aligned} \text{gpm} = \text{gallons per minute} &= \frac{1728v (\text{Heat requirements in Mbh})}{0.231 \times 60(h_{fs} - h_{fr})} \\ &= \frac{124v (\text{Heat requirements in Mbh})}{(h_{fs} - h_{fr})} \end{aligned} \quad (7-11)$$

Mbh = 1000 Btu per hr;

h_{fs} = enthalpy of liquid at heater outlet, Btu per pound;

h_{fr} = enthalpy of liquid at return to heater, Btu per pound;

v = volume of 1 lb liquid at mean temperature, cubic feet;

W = pounds per hour;

gpm = gallons per minute.

For a mean temperature of 170° F this reduces to

$$\text{gpm} = \frac{2.05 \text{ Heat requirements in Mbh}}{(h_{fs} - h_{fr})} \quad (7-12)$$

Since $h_{fs} - h_{fr}$ is equal to $t_s - t_r$ with close approximation, the formula above for a range from 180° F to 160° F or 20° F becomes

$$\text{gpm} = 0.102 \text{ Heat demand in Mbh} \quad (7-13)$$

This is the total amount of water which must be circulated with a 20° drop in temperature, no matter what arrangement of the piping or radiators is used. The same equation gives the water needed by any radiator if one could assume this same temperature drop in it. This cannot be done except in two-pipe systems with covered risers, as the

hot-water system is really a network of conduits in which there must be a balance between the heat delivery from the radiators, the three temperatures at the radiators and the water used by them, as well as a balance in the friction drops through each parallel circuit. After this balance, equations (7-10), (7-11) or (7-12) may be used to determine the amount of water.

To solve the problem of friction many experimenters have worked on the resistances to water flow, and their results have been embodied in the analysis which follows.

The head causing flow may be due to the gravity action of the water columns or to this combined with that added from a pump. In these systems the velocities are so low that, in the lengths of pipe used, the friction loss is small, and for that reason it is reported in thousandths of an inch, milli-inches, in some unit of length. The velocities are so small that they are expressed in inches per second.

The gravity circulation head produced by the difference in the weights of the two columns is determined by considering densities.

Pressure difference for columns of L feet height in pounds per square foot = $L(w_c - w_h)$ (7-14)

Head in feet of water of mean density = $\frac{\text{Pressure}}{\text{Mean density}}$

$$= \frac{L(w_c - w_h)}{\frac{w_c + w_h}{2}} = \frac{2L(w_c - w_h)}{w_c + w_h}$$

$$\text{Head in milli-inches} = 24\,000 \frac{L(w_c - w_h)}{(w_c + w_h)} = 24\,000 \frac{L(v_h - v_c)}{v_h + v_c} \quad (7-15)$$

w_c = weight of 1 cu ft at low temperature;

w_h = weight of 1 cu ft at high temperature;

v_c = volume of 1 lb at low temperature;

v_h = volume of 1 lb at high temperature;

L = height of columns in feet.

The following table gives the head in milli-inches of water at mean temperature produced by a height of 1 ft with different temperatures in the two legs of the system.

TABLE 7-5

CIRCULATION HEAD IN MILLI-INCHES OF WATER OF MEAN TEMPERATURE PRODUCED BY ONE FOOT ELEVATION

		Temperature of Supply—Degrees F															
Temperature of Return, Degrees F		90	100	110	120	130	140	150	160	170	180	190	200	210	220	230	240
	80	21	41	70	99	130	163										
	90	24	48	78	109	143	178									
	100	27	55	86	119	156	193								
	110	29	61	93	129	168	207							
	120	31	65	100	138	179	221						
	130	33	69	107	147	190	235					
	140	35	73	114	156	201	248				
	150	39	77	121	165	212	261			
	160	41	82	127	174	223	274		
	170	43	86	134	183	234	286	
	180	45	90	140	192	244	296
	190	47	94	147	200	255
	200	49	98	151	208
	210	50	102	157
	220	52	106
	230	53

The head produced by the pump must be added to the gravity head in installations using forced circulation.

The circulation head is used in causing flow against friction losses. The friction loss in pipes is given by

$$F = 4f \frac{L V^2}{d 2g} \quad (7-16)$$

where f , the friction factor, depends upon the Reynolds number,

$$\frac{dVw}{\mu} \quad \text{or} \quad \frac{dG}{\mu}.$$

F = head lost in L feet of length in feet of head of material flowing;

V = velocity in feet per second;

d = diameter in feet;

w = density of fluid, pounds per cubic foot;

μ = absolute viscosity pounds per second-foot;

G = pounds of material flowing per second per square foot of pipe section;

g = acceleration of gravity, feet per second per second.

The quantity V may be eliminated in terms of the heat per hour and the cross-sectional area of the pipe.

$$3600VAw = W \text{ pounds per hour}$$

$$\frac{3600VAw(h_{fs} - h_{fr})}{1000} = \text{Mbh} = \frac{3600 \times Vaw(h_{fs} - h_{fr})}{144\,000} \quad (7-17)$$

A = area of interior of pipe in square feet;

a = area of interior of pipe in square inches.

For a drop in temperature of 10° , $(h_{fs} - h_{fr})$ will equal almost 10 Btu, and in hot-water heating with 170°F for the mean water temperature, w is 60.8 lb per cu ft. This gives

$$V = \frac{\text{Mbh}}{2189A} = \frac{6.58 \times 10^{-2} \text{ Mbh}}{a} \quad (7-18)$$

hence

$$\frac{F}{L} = \frac{4f}{d} \frac{(\text{Mbh})^2}{(2189)^2 A^2 2g} = \frac{f(\text{Mbh})^2}{4.78 \times 10^7 d^5} \quad (7-19)$$

If d is measured in inches and F/L in milli-inches per foot with a 10°F temperature drop, the equation becomes

$$\text{Friction loss in milli-inches per foot} = \frac{62.7f(\text{Mbh})^2}{d^5} \quad (7-20)$$

d = internal diameter of pipe in inches.

The value of f depends on the Reynolds number, $\frac{dVw}{\mu}$.

d = diameter of pipe in feet;

V = velocity in feet per second;

w = weight of 1 cu ft in pounds;

μ = absolute viscosity in 6.72×10^{-4} centipoise, pounds per second foot.

The Reynolds numbers have been computed for definite pipe sizes and velocities with water at definite temperatures, and from these the value of f has been selected from the curves given by McAdam in his *Heat Transfer*. The values of f were computed for four different velocities and for five pipe sizes with temperatures of 150°F and 190°F . The variation in f due to velocity change was 100% of the smallest value, and a similar change was found with different pipes at the same velocity, but with a change of 40°F in water temperatures the variation of f was 5% and less, except in small pipes at low velocities where the difference

was 10%. For this reason the friction loss in hot-water pipes carrying different heat demands with a drop of 10°F was computed at a mean temperature of 170°F , and the results have been plotted in logarithmic

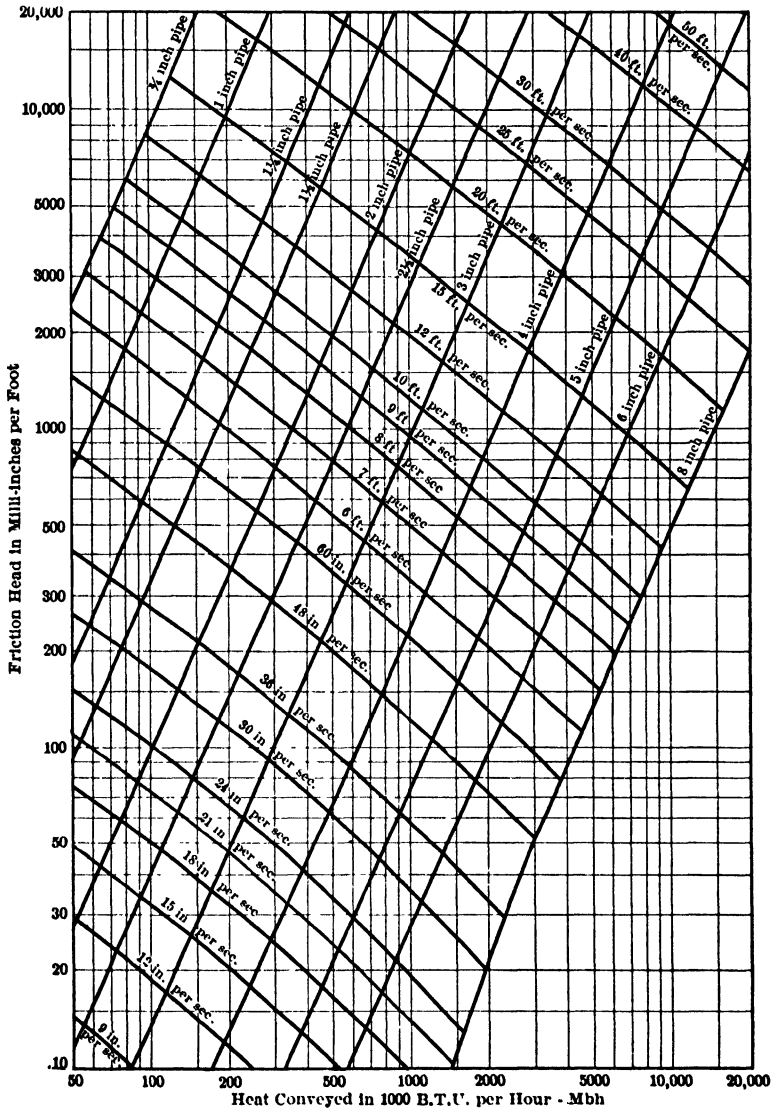


FIG. 138.—Heat Capacity, Friction Loss and Velocity for Pipes Supplying Water with 10°F Decrease of Temperature in Radiators.

form for greater spread of values as has been done by Giesecke using a different formulation. The curves of Fig. 138 indicate resistance, capaci-

ties and velocities of pipe for large installations; Fig. 139 is given for small systems.

The friction in elbows and other fittings has been studied for many years in a number of investigations in the fields of water supply and of hot-water heating. The extensive work done by F. E. Giesecke and his associates at the University of Texas has been especially directed to the field of hot-water heating.

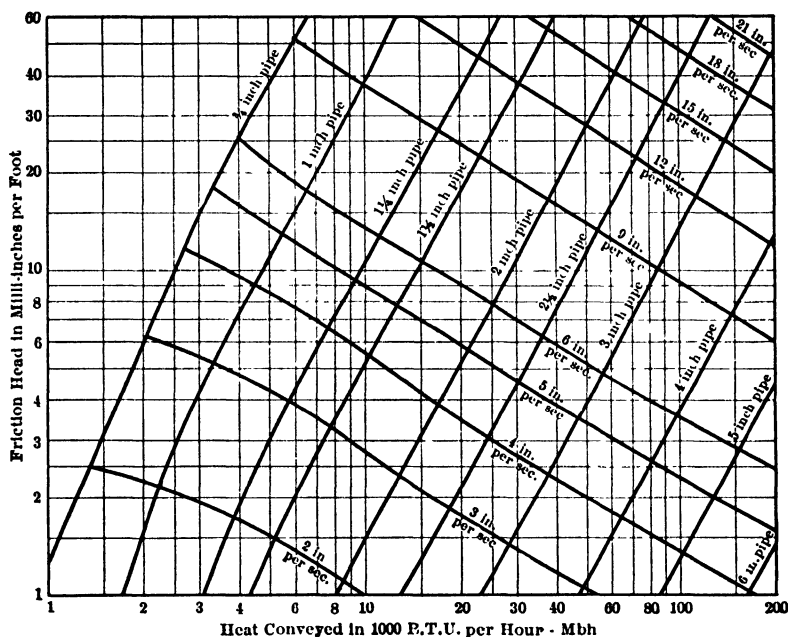


Fig. 139.—Heat Capacity, Friction and Velocity in Pipes Supplying Water with 10° F Decrease of Temperature in Radiator.

These experiments have shown that the friction and impact loss in elbows depends on the velocity and diameter so that the value of c in

$$h_1 = c \frac{V^2}{2g}$$

for the loss in elbows with a velocity V was not a true constant but varied from a value less than unity to one above unity, 1.3. Because of that slight variation and because of the variation in elbows, it was suggested in the 1935 Guide that the coefficient be taken as unity.

Since the approximate friction drop in pipe is

$$0.02 \frac{L}{d} \frac{V^2}{2g}$$

the friction loss in an elbow is considered to be equivalent to that in 50 diameters of pipe. When V is given in inches per second the elbow loss becomes

$$\begin{aligned}\text{Loss in elbow in milli-inches of water} &= \frac{12\,000}{144} \frac{V^2}{2g} \\ &= 1.3V^2\end{aligned}\quad (7-21)$$

V = velocity in inches per second

TABLE 7-6
VELOCITY HEAD OR FRICTION LOSS IN ELBOW
 V in Inches per Second

V	Mi	V	Mi	V	Mi	V	Mi
2	5	9	105	24	748	72	6 720
3	12	12	187	30	1168	84	9 150
4	21	15	291	36	1680	96	11 950
5	33	18	420	48	2990	108	15 130
6	47	21	572	60	4670	120	18 680

Experiments have been made by Giesecke and Badgett * to determine the resistance of tees when varying amounts of the total water have been taken from or given to the branch or run.

The loss in head at a tee in the circuit of the branch carrying water from the run of a tee is given as

$$\begin{aligned}\text{Loss in branch circuit} &= \frac{V^2}{2g} \text{ for branch} \div \left(\frac{\text{Percentage taken off}}{100} \right)^2 \\ &= \frac{\text{Loss in branch ell}}{\left(\frac{\text{Percentage off}}{100} \right)^2}\end{aligned}\quad (7-22)$$

The percentage taken off means the percentage of the water entering the run which is taken off by the branch. This formula can be used to 60% diversion. For 70% the multiplier of $\frac{V^2}{2g}$, or the loss in the branch ell, is 2.45; for 80%, 2.20; for 90%, 1.95, and for 100%, when a plugged tee is used for an ell, 1.9.

* Friction Head in Cast Iron Tees, by Giesecke and Badgett, Trans. A.S.H.V.E., Vol. 37, 1931; Vol. 38, 1932.

The loss at a tee in the circuit of the run for various percentages of water entering the run which continue through is given as that of a number of ells carrying this percentage of water.

TABLE 7-7

EQUIVALENT ELLS IN RUN CIRCUIT OF TEE

Percentage of water continuing.....	100	80	60	50	40	30
Number of ells on run outlet	0.06	0.12	0.30	0.58	1.2	3.4

The loss at a tee in the branch circuit when a given percentage of the water discharged from the run enters from the branch is expressed as that of a number of ells carrying this percentage of water.

TABLE 7-8

EQUIVALENT ELLS IN BRANCH CIRCUIT DISCHARGING INTO TEE

Percentage entering from branch.....	100	70	50	30	20	10
Number of branch ells.....	1.8	2.0	2.7	6.6	16.0	57.0

The loss at a tee in the run circuit when a given percentage of the water leaving the run enters from the other side of the run is expressed as that of the number of ells carrying this percentage of water.

TABLE 7-9

EQUIVALENT ELLS IN RUN CIRCUIT DISCHARGING THROUGH RUN

Percentage of water entering run.....	100	80	60	40	20	10
Number of entering run ells.....	0.08	1.2	1.7	3.2	11	70

The loss at a tee when water enters from a branch and divides into two circuits from the ends of the run is expressed for either circuit as that of a number of ells carrying the percentage of water given in Table 7-10.

TABLE 7-10

EQUIVALENT ELLS IN SPLIT CIRCUIT FROM BRANCH

Percentage of branch water leaving run.....	100	80	60	40	20	10
Number of run ells.....	1.8	1.9	2.4	4.6	17	75

The loss at a tee when water enters from each side of the run and discharges through the branch is expressed, for either circuit, as that of a number of ells carrying the percentage of water given in Table 7-11.

TABLE 7-11

EQUIVALENT ELLS IN RUN CIRCUITS DISCHARGING INTO BRANCH

Percentage of branch water entering run....	100	80	60	40	20	15
Number of entering run elbows.....	1.9	2.4	3.3	6.7	24	50

For other fittings the Guide and others have given the resistance in terms of that of a number of ells. From these the following table has been prepared.

TABLE 7-12

RESISTANCE IN FITTINGS IN TERMS OF RESISTANCE OF
NUMBER OF ELBOWS OF SAME SIZE

Blanked tee used as ell (1.9 to 2.5)	2.0 ell
Long sweep elbows	0.5 to 0.6 ell
Pipe bends	0.1 to 0.5 ell
45° elbows	0.6 to 0.7 ell
Cast-iron bends	1.0 ell
Gate valves, screwed	0.5 ell
Gate valves, flanged	0.0 ell
Globe valve	10 to 20 ell
Angle valve	9 ell
Angle radiator valve	2 ell
Radiators	3 ell
Heaters	2.5 to 3.5 ell

Two-Pipe System Reversed Return. The resistance of the various elements of the system being known, an examination must be made of the various circuits so as to balance the available head and the resistance to flow of the required water.

Two circuits of the building of Chapter V will be considered to illustrate the methods of selecting pipe sizes, one for riser *A* and one for riser *H*, Fig. 137. Sketches are made giving the various lengths, heights, and heat quantities at various points, Fig. 140, using the cellar plan, Fig. 141, to obtain dimensions. This cellar plan is selected to fit the development drawing of the building, Fig. 106, using a reversed flow two-pipe system.

In this design it is first assumed that the water is delivered at 180° F and returned at 160° F producing 82 mi per ft of elevation.

The head from the middle height of the water in the heater to inlet of the second-story radiator is 18 ft, to the inlet of the first-story radiator 8 ft and to the flow line 4½ ft.

The total head causing flow to the first-floor radiator on this circuit is 8×82 or 656 mi, and that to the upper radiator is 1476 mi. Since this additional head from the upper radiator will cause the flow to by-pass the first radiator (inducing reversed

flow in it) the head due to the 10 ft difference in elevation will be cared for by friction from an orifice.

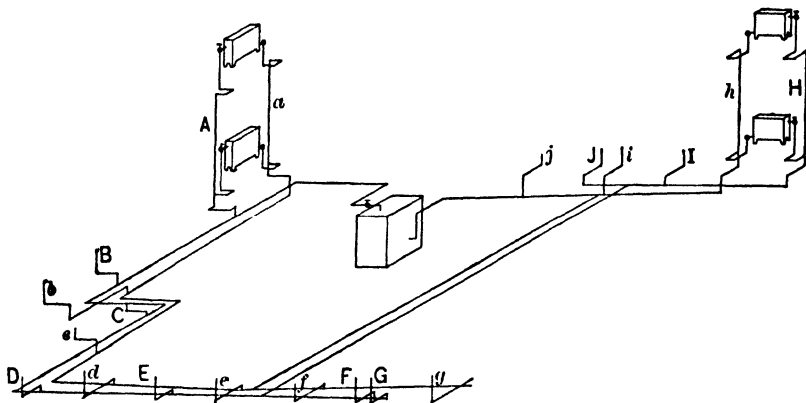


FIG. 140.—Oblique Drawing of Circuits of Risers A and H.

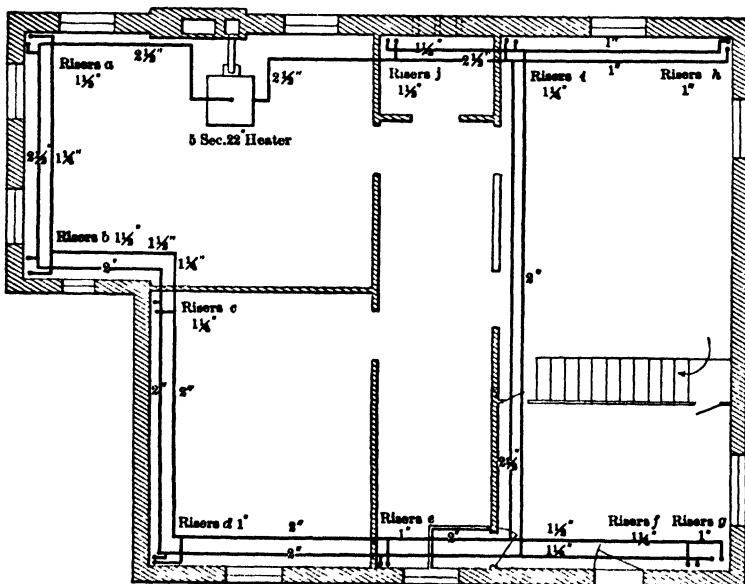


FIG. 141.—Flow Mains in Cellar.

The data taken from Figs. 137, 140 and 141 for the circuits to riser A, using Tables 7-7 to 7-12 are arranged in tabular form. For each riser the data may be arranged in a similar manner as that of Table 7-13 for Riser A.

TABLE 7-13
CIRCUIT OF RISER A
SUPPLY

		Equiva- lent Ells		Equiva- lent Ells
Heater to A	Mbh..... 116.1	3	Tee..... 15% ($\frac{1}{4}$)	11
	Heater.....		Ells.....	4
	Pipe..... 18 ft		Tee.....	3
	Ells.....		Ells.....	3
Branch at A	Mbh..... 18.0	4	Pipe..... 6 ft	
	Pipe..... 4 ft		Radiator valve.....	2
			Radiator.....	1 $\frac{1}{2}$

Note: Flanged gate valve used.

Total Pipe..... 28 ft

Total equivalent ells..... 31 $\frac{1}{2}$

RETURNS

		Equiv- alent Ells			Equiv- alent Ells
Branch at A	Tee 55%	2.7	To E	Mbh 62.0	1
	Pipe 4 ft			Pipe 14 ft	
	Radiator			Tee 9%	
	Radiator valve			Mbh 67.8	
To B	Ells	6	Junc. 1	Pipe 1 ft	2.4
	Mbh 18.0			Tee 80%	
	Pipe 14 ft			Mbh 82.7	
	Tee 50%			Pipe 30 ft	
To C	Mbh 36.0	2	To J	Tee 90%	2
	Pipe 10 ft			Mbh 101.3	
	Ells			Pipe 8 ft	
	Tee 33%			Tee 90%	
To D	Mbh 54.4	2	To Heater	Tee 13%	1
	Pipe 15 ft			Mbh 116.1	
	Ells			Pipe 16 ft	
	Tee 12%			Ells	

Total pipe..... 112.0

Total equivalent ells..... 31.6

The total pipe length of this circuit is 140 ft, and there are 63 equivalent elbows. As a first approximation one-half of the friction will be assumed to be used up in elbows giving 2.2 mi per ft for the pipe friction. The drop in temperature being

20° F in place of 10° as plotted on the lines of Figs. 138, 139, only $\frac{1}{2}$ of the Mbh values will be used in selecting the pipe sizes. The preliminary sizes are then found as follows:

TABLE 7-14

PIPE SIZES

Supply		Return			
Heater to A	4"	Branch	2"	E to junction	3"
Branch A	2"	A to B	2"	Junction to junction..	3"
		B to C	2½"	Junction to I	4"
		C to D	3"	I to J	4"
		D to E	3"	J to heater	4"

These pipe sizes are so large that either a circulating pump should be added or the temperature difference should be increased. If the temperatures are now made 190° and 150°, so that the mean temperature remains 170° F, the head per foot would be 165 mi and the total head to the first-floor radiators would be 1320 mi. One-half of this for pipe friction gives 4.3 mi per ft, using 4.5 mi per ft. The sizes are then selected from the curves at $\frac{1}{4}$ of the Mbh actually needed.

TABLE 7-15

PIPE SIZES

Supply		Return			
Heater to A	2½"	Branch	1½"	E to junction	2"
Branch	1½"	A to B	1½"	Junction to junction..	2½" (2")
		B to C	1½"	Junction to I	2½" (2")
		C to D	2"	I to J	2½" (2")
		D to E	2"	J to heater	2½"

The pipe sizes being known, the actual total loss of head is computed including that from the fittings. The loss in equivalent elbows is determined as $1.3V^2$, the velocities being taken from Figs. 138 and 139.

TABLE 7-16
CIRCUIT FOR RISER A—RESISTANCE IN MI.

Supply				Return			
To A	Heater.....	3×30	90	Branch	Radiator...	1½ × 8	12
2½"	Pipe.....	18× 4.2	75	1¼"	Radiator		
	Ells.....	4×30	120		valve....	2 × 8	16
Branch	Pipe.....	14× 2.5	25		Pipe.....	4 × 2.5	10
1¼"	Tee.....	11× 8	88		Ells.....	6 × 8	48
	Ells.....	4× 8	32		Tee.....	2.7× 8	22
	Tee.....	3× 8	24	To B	Pipe.....	14 × 2.5	35
	Ells.....	3× 8	24	1¼"	Ells.....	2 × 8.0	16
	Radiator			To C	Pipe.....	10 × 4.3	43
	valve.....	2× 8	16	1½"	Ells.....	4 × 16	64
	Radiator.....	1½ × 8	12	To D	Pipe.....	15 × 3.0	45
	Total		506	2"	Ells.....	2 × 16	32
				To E	Pipe.....	14 × 3.5	49
				2"	Ells.....	1 × 18	18
				To Junc.	Pipe.....	1 × 3.6	4
				2"	Ells.....	2.4×18	42
				To Junc.	Pipe.....	30 × 2.4	72
				2½"	Ells.....	2 × 15	30
				To J	Pipe.....	8 × 3.1	25
				2½"	Ells.....	3 × 21	63
				To Heat	Pipe.....	16 × 4.2	67
				2½"	Ells.....	3 × 30	90
				Total....			803

Grand total 1309 mi.

The amount available is 1320 mi. This is sufficiently close.

The riser A from first to second floor has 1650 mi available for 7.9 Mbh through 30 ft of pipe and 22 elbows. If ¾ in. pipe is used for this, the resistance is

$$\text{Pipe } 30 \times 6.2 = 186$$

$$\text{Ells } 22 \times 12 = 264$$

$$450 \text{ mi}$$

The remaining 1200 mi must be cared for by an orifice in the valve or in a union. The velocity is 3 in. per sec.

$$1200 = c' \times 9$$

$$c' = 134$$

$$\frac{A_o}{A_p} = 0.14$$

$$d_o = 0.37 d_p = 0.37 \times 0.82 = 0.306 \text{ in.}$$

For the riser H the sizes of the return flow line have been fixed except for the short runout to the corner, and it will be necessary to compute the sizes of this return and those for the supply flow line to this point. The sizes of the supply line will be selected on the assumption of 4.5 mi per ft with a 40° temperature drop. This means that points will be selected from the curves by using ¼ of the Mbh of those parts of the circuit.

TABLE 7-17
CIRCUIT FOR RISER *H*

Supply				Return			
Heater	Mbh.....	116.1		Riser	Mbh.....	9.4	
to <i>A</i>	Heater.....	3×30	90	1"	Pipe.....	4×2.4	10
2½"	Pipe.....	18×4.2	76		Radiator.....	1½×6	9
	Ells.....	4×30	120		Ells Equivalent	10×6	60
<i>A</i> to <i>B</i>	Mbh.....	98.1		<i>H</i> to <i>J2</i>	Mbh.....	9.4	
2½"	Pipe.....	14×3.0	42		Pipe.....	12×2.4	29
	Tee.....	2×21	42		Tee.....	2×6	12
<i>B</i> to <i>C</i>	Mbh.....	80.1		<i>J2</i> to <i>I</i>	Mbh.....	92.0	
2"	Pipe.....	10×5.2	52	2"	Pipe.....	1×7	7
	Ells.....	2×18	36		Tee.....	2×40	80
	Tee.....	2×18	36	<i>I</i> to <i>J</i>	Mbh.....	101.3	
<i>C</i> to <i>D</i>	Mbh.....	61.7		2"	Pipe.....	8×8	64
2"	Pipe.....	17×3.4	58		Tee.....	2×46	92
	Ells.....	1×18	18	<i>J</i> to	Mbh.....	116.1	
	Tee.....	2×18	36	heater	Pipe.....	20×4.2	84
<i>D</i> to <i>E</i>	Mbh.....	54.1		2½"	Ells.....	4×30	120
2"	Pipe.....	14×2.6	36		Total.....		567
	Tee.....	2×16	32				
<i>E</i> to <i>J1</i>	Mbh.....	48.3					
2"	Pipe.....	8×1.7	14				
	Tee.....	2×10	20				
<i>J1</i> to <i>J2</i>	Mbh.....	33.4					
1½"	Pipe.....	30×4.0	120				
	Tee.....	2×15	30				
<i>J2</i> to <i>H</i>	Mbh.....	9.4					
1"	Pipe.....	12×2.4	29				
	Ells.....	5×6	30				
Risers	Mbh.....	9.4					
1"	Pipe.....	6×2.4	14				
	Ells.....	3×6	18				
	Radiator valve	2×6	12				
	Radiator....	1½×6	9				
	Total.....		970				

The total 1537 mi is too great. The supply line to *B* and to *D* will be made 2½ in., and that from *J-1* to *J-2*, 2 in. These changes will make the total friction 1370 mi, which is near the amount available.

The riser *H* to the second floor will be made ¾ in. and an orifice will be installed as for riser *A*.

As the sizes of the flow lines have been determined it is possible to find the difference between the pressures at the supply and return runouts of the various risers and at the branch mains to risers *F* and *G* and so determine their sizes to care for the heat delivery.

Single-Pipe System. If a single-pipe system is installed in this building the flow main will be carried completely around the cellar requiring 157 ft of pipe and 12

elbows. The lengths are so short in this installation that expansion will not be great and single ells may be used at all angles of the flow line.

The head causing flow in the main with the drop from 190 to 150 is $4\frac{1}{2} \times 165$ or 743 mi, and if 4 mi per ft be assumed the main will have to be slightly larger than $2\frac{1}{2}$ in. to carry 116.1 Mbh with a 40° drop in water temperature. The velocity is almost 4.8 in. per sec giving an elbow loss of 30 mi.

$$\begin{aligned} 2\frac{1}{2} \text{ in. pipe } 157 \times 4.3 &= 755 \text{ mi} \\ 2\frac{1}{2} \text{ in. ells } 12 \times 30 &= 360 \end{aligned}$$

1115 mi

As this requires more than the available head the water temperature range may be changed to 192° F and 148° F which will give a circulating head of 810 mi for a resistance of 502 mi in the pipe and 266 mi in the elbows with 4.4 in. per sec velocity, a total of 768 mi. This resistance will be reduced slightly as at each riser there is a shunt circuit which reduces the friction drop in the shunt lengths between the run-outs to the supply risers and those of the return risers. These lengths vary from a foot to 4 ft, so that for the ten risers there may be a reduction of 20 mi or 2% in the friction loss.

The temperature of the water will gradually decrease as it passes around the flow line starting at 192.0 for riser A and ending at 148.0 at the return from riser J. The average temperatures at each riser inlet will be found as follows

$$t_x = 192 - \frac{44(\text{Mbh to riser } x)}{116.1} \quad (7-23)$$

The following table gives the temperatures at entrance to each riser, the mean temperature, assuming the outlet temperature to be 148° F, and the radiation factors to redetermine the surface of the radiators on each riser. (With the single-pipe system radiators near the return point must be increased in size because of the lower mean water temperature.)

TABLE 7-18
SINGLE-PIPE SYSTEM

Risers	Riser Cap. Mbh	2nd Floor Mbh	Sum	Temper- ature Inlet	Mean Temper- ature	Drop in Temper- ature	Circu- lation Head Mi in 1 Ft	Radi- ation Factor	Mbh for 10° Drop	
									Branch	2nd Floor
A	18 0	7.9	18.0	192.0	170.0	44.0	180	0.617	4.1	1.8
B	18 0	7.9	36.0	185.2	166.8	37.2	150	0.591	4 8	2.1
C	18.4	10.8	54.4	178.5	163.3	30 5	120	0.562	6.1	3 5
D	7 6	0	62.0	171.5	159 7	23.5	90	0.534	3.2	
E	5.8	3 6	67.8	168.8	158.4	20.8	80	0.523	2.8	1.7
F	10.7	4 2	78.5	166.6	157.3	18.6	63	0.514	5 7	2.2
G	4 2	4 2	82.7	163.0	155.5	15.0	48	0.500	2.8	2.8
H	9 3	4 3	92.0	161.0	154.5	13.0	45	0.492	7 2	3.3
I	9.3	4.3	101.3	157.6	152.8	9.6	39	0.486	9.7	4.5
J	14.8	2.6	116.1	154.0	151.0	6.0	24	0.470	25.0	4.3

The drops in temperature in the radiators have been computed in order to reduce the values of Mbh to the Mbh for equal flow with a 10° drop in temperature so as to use the chart, Fig. 139. The riser sizes may be selected and velocity of flow for any given pipe size is taken from the chart as this is needed for elbow resistance and orifice determination. The gravity circulation head per foot of height must be obtained to determine the head for the first-floor radiators from $3\frac{1}{2}$ ft above the flow main and for the second floor, from $13\frac{1}{2}$ ft. The riser circuits will include a tee equivalent to 40 branch ells for 15% diversion, but since the velocities in the branches are low this may be taken as 10 ells. There are also 4 ells and 3 ft of pipe for the runout. The supply riser will have $1\frac{1}{2}$ ft to the first-floor radiators and 10 additional feet to the second-floor radiators. The radiator connections will include one tee or ell, 3 ells, 4 ft of pipe and a radiator valve. For the returns there will be a valve, 2 ft of pipe, 3 ells, an ell or tee for the first-floor radiator connection, 10 ft and $1\frac{1}{2}$ ft of risers and 4 ft of pipe with a tee and 4 ells for the runout, as well as the tee resistance in the line. As seen earlier, the risers to the second-floor radiators will be controlled by orifices and the sizes of the risers to the first-floor radiators will be determined by friction drop unless the required pipe size is less than $\frac{3}{4}$ in, in which case an orifice will be used in a $\frac{3}{4}$ in. pipe. The circuits to the first-floor radiator include 4 tees or 23 equivalent ells, 7 equivalent ells for radiator and valves, 12 ells and 12 ft of pipe. For riser A the resistance for a 1 in. pipe carrying 4.1 Mbh is 7.1 mi, and the water has a velocity of 3.6 ft per sec for which the velocity head is 16.7 mi. The total of this is 784 mi. The head available from $3\frac{1}{2}$ ft (3.5×180) is 630 mi. The pipe must be enlarged and $1\frac{1}{4}$ in. line must be used to the first floor. This uses only 318 mi so that an orifice must be used.

In the same manner the other risers are worked out and entered in Table 7-19.

Orifice Size. To show the method of selecting orifices riser C will be used. In this the loss in a $1\frac{1}{2}$ in. pipe for 6.1 Mbh is 2.3 mi per ft and the velocity is 2.5 in. per sec.

$$\begin{array}{rcl} \text{Pipe } 12 \times 2.3 & = & 28 \\ \text{Elbow } 42 \times 6.6 & = & 277 \\ \hline \text{Total} & & 305 \text{ mi} \end{array}$$

Amount available 420
Excess head 115 mi

Since velocity is 2.3 in. per sec

$$115 = c' \times 5.3$$

$$c' = 22.0$$

From curve, Fig. 134,

$$c' = 22.0 \text{ gives}$$

$$\frac{A}{A_p} = 0.375 \text{ and } d_o = \sqrt{0.375} \times 1.61 = 0.99 \text{ in.}$$

For the upper section of the riser carrying the equivalent 3.5 Mbh at 10° , there is available 1200 mi of circulation head to be used in 26 ft of pipe, and 19 equivalent ells. The riser must be larger than $\frac{3}{4}$ in. If these riser pipes be made 1 in. the friction at a velocity 3.2 in. per sec is

$$\begin{array}{rcl} \text{Pipe } 26 \times 6 & = & 156 \\ \text{Ells } 19 \times 13.5 & = & 257 \\ \hline \text{Total} & & 413 \end{array}$$

TABLE 7-19

HOT WATER RISER SIZES AND RADIATOR SURFACES—
SINGLE PIPE

Riser	Radiator	Surface	Mbh Actual	Equivalent 10° Mbh	Mi 1st	Size Riser	Size Used	Equivalent Mbh Top
<i>A</i>	101a 201a	68 53	10 1 7 9	4.1	630	1+	1 $\frac{1}{4}$	1.75
<i>B</i>	101b 201b	71 55	10.1 7 9	4.8	525	1+	1 $\frac{1}{4}$	2.10
<i>C</i>	102a 202a	56 80	7.6 10 8	6 1	420	1 $\frac{1}{4}$ +	1 $\frac{1}{2}$	3.5
<i>D</i>	102b	59	7 6	3 2	315	$\frac{3}{4}$	1	3.2
<i>E</i>	103a 203a	18 34	2.2 3 6	2 8	280	1+	1 $\frac{1}{4}$	1.7
<i>F</i>	104a 204a	53 34	6 5 4 2	5 7	220	1 $\frac{1}{4}$ +	1 $\frac{1}{2}$	2.2
<i>G</i>	204b	35	4 2	2 8	178	1+	1 $\frac{1}{4}$	2.8
<i>H</i>	105a 205a	43 37	5 0 4 3	7 2	158	2+	2	3.3
<i>I</i>	105b 205b	43 37	5 0 4 3	9 7	137	2+	2 $\frac{1}{2}$	4.4
<i>J</i>	106a 206a	108 26	12 2 2.7	24.8	84	2 $\frac{1}{2}$	4	4 6
Total	910	71.2				

Excess circulation head 787 mi.

$$c' = \frac{787}{10.2} = 77.0$$

$$\frac{A_o}{A_p} = 0.17$$

$$d_o = \sqrt{0.17} \times 1.05 = 0.434 \text{ in.}$$

Forced Circulation. If it is desired to reduce the pipe sizes in either of the solutions above, a small circulating centrifugal pump would be added to the system. Thus in the double-pipe system a pump might be added to increase the operating head by 2640 mi. This would give at least 13.5 mi per ft of length and give the following sizes, which may be compared with those in Table 7-15.

TABLE 7-20
HOT WATER PIPING WITH FORCED CIRCULATION

Supply	Return	
Heater to A 2"	Branch 1"	E to junction 1½"
Branch 1"	A to B 1"	Junction to junction 2"
	B to C 1¼"	Junction to I 2"
	C to D 1½"	I to J 2"
	D to E 1½"	J to heater 2"

The water circulated per hour with a 40° drop for 116.1 Mbh is given as

$$\text{Lb per hr} = \frac{1000 \times 116.1}{40} = 2905 \text{ lb}$$

$$\text{Lb per min} = 48.4 \text{ lb}$$

The horsepower supplied to the circulator to produce this head of 2640 mi or $\frac{2.6}{12}$ ft with 50% efficiency of pump and motor would be

$$\frac{48.4 \times 2.64}{33\,000 \times 12 \times 0.50} = 0.0006 \text{ hp}$$

This low power suggests the increase of head and the further reduction of pipe sizes. Forced circulation requires small power and reduces the sizes of pipes so much that it may be advisable. The use of hot water for district heating with forced circulation has many possibilities.

Shunted Circuits. The shunted circuit mentioned for the one-pipe system will be examined for circulation without the use of an orifice. Suppose that for riser *D* the length between the two runouts on the 2½-in. flow line is 36 in. and there is normally a drop of 3.4 mi per ft in the main. If the arrangement of the branch is as shown in Fig. 142, there would be a drop in the flow line between runoffs of

$$3.4 \times 3 \times \left(\frac{108.6}{116.2} \right)^2 = 8.9 \text{ mi.}$$

The resistance through the radiator circuit is due to 12 ft of pipe and 36 equivalent elbows. The circulation head of $3\frac{1}{2}$ ft of water is

$$h = 3\frac{1}{2} \times 90 = 315 \text{ mi}$$

This with the 8.9 mi in the pipe gives a total drop through the radiator circuit of 323.9 mi.

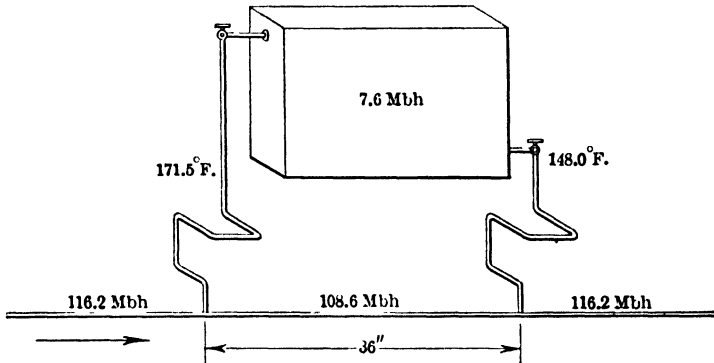


FIG. 142.—Shunted Circuits.

The temperature drop through this radiator is 23.5°F , as given in the table, hence to use the diagrams for a 10° drop in temperature the Mbh is given by

$$\frac{7.6}{2.35} = 3.2 \text{ Mbh (for use on chart)}$$

At this value of Mbh the resistance of 1 ft of 1-in. pipe is 5.0 mi and the velocity is 2.9 in. per sec, for which the velocity head is 11 mi. For a $1\frac{1}{4}$ -in. pipe and elbow with a velocity of 1.3 in. per sec the values are 1.0 and 2.2 mi, respectively. The 1-in. pipe would require 431 mi and the $1\frac{1}{4}$ -in. pipe 91 mi.

The $1\frac{1}{4}$ -in. pipe must be used, and as it does not utilize the full circulation head more water will flow so that the outlet temperature will be higher than 148.0°F and will be assumed to be 153.0°F . The drop in temperature is 18.5 and the circulation head is 73 mi per ft or 256 mi for $3\frac{1}{2}$ ft. The Mbh to be used on the chart is now 4.1. This gives a loss per foot of 2.0 mi and a velocity of 2.2 in. per sec or a velocity head of 6.3 mi.

Total circulating head	= 256.0
Pipe friction 12×2.0	= 24
Equivalent ells 36×6.3	= 227
	—
Total	= 251

The assumed temperature is sufficiently close. The radiator surface will be changed from that in the table because of the slightly higher mean water temperature (159.7 is changed to 162.2).

$$\text{Surface} = 59 \times \frac{0.534}{0.554} = 57 \text{ sq ft}$$

It is to be realized that although more water is sent through the radiator it has given up the same amount of heat and the temperature in the flow main after the water mixes with that of the main will be that given in the table as entering the next riser *E*. This way of handling shunted circuits is time-consuming but indicates the method to be pursued if it is desired to eliminate orifices.

The pipes in the hot-water system should be installed so that the slight expansion occurring can take place without binding and there should be a slight pitch to permit drainage when necessary. There should be no pockets in which water may remain when the system is to be drained for vacancy during the winter. Although a rising pitch in the direction of flow in the flow main and a falling pitch in the return main may be advantageous in starting flow on first heating, this is not necessary after flow is once started as the system is completely filled with water and the circulating force causes flow in the direction of this pressure difference regardless of pitch.

PROBLEMS

1. Find the sizes of the hot water radiators for rooms given on the general data sheet, Fig. 96, for hot water supply at 200° F and a return at 170° F. How large should pipes be for 3 mi per ft.? What would be the total drop for first-floor radiator on riser *B* in this case? Is there enough gravity head for this?

2. How much water must be circulated in problem 1? Lay out the flow lines in this basement with a pump which produces 5 in. of water pressure in addition to the gravity circulation head. What power would be consumed by this pump with 50% over-all efficiency?

3. A 1 in. line supplying 20 Mbh through 30 ft and 18 equivalent ells has water temperatures of 180° F and 150° F with a gravity head of 20 ft. How large an orifice should be used in this line?

4. A building has 700 sq ft of radiation. The water of the system may vary from 90° F to 200° F. Compute the size of the expansion tank. Is Table 7-3 conservative?

5. Determine the elbow equivalents of the circuit of Fig. 142.

6. Design a single-pipe system for building of Fig. 96 with a temperature variation from 210° F to 150° F.

7. Compute the flow to the shunt circuit of riser *D* in problem 6.

CHAPTER VIII

INDIRECT STEAM HEATING—AIR FLOW—GRAVITY SYSTEMS

Indirect steam heating refers to the supply of heat for the room losses by means of air at such a temperature that in cooling to that of the room it liberates the necessary energy. There are three general methods of indirect steam heating as mentioned in Chapter I. In the natural draft or gravity method, heating units of the pin or fin type are placed in enclosures at the bases of flues or risers leading to the rooms to be heated while inlets supply air to the enclosures from the outside or from recirculating ducts. In the plenum or forced mechanical method, air from the outside or recirculation air is forced over extended heating surface by the action of fans and delivered into ducts and risers to the various rooms or unit heaters with their fans and radiators are placed at suitable points in each room to supply the heated air. In the third, or vacuum system, air is drawn from the rooms by fans while the air supply, brought to the room by ducts, is caused to pass over heaters located at some central point.

In all these systems the heat given up by the air for a room on cooling from its initial temperature t_i to the room temperature is expressed by

$$q_{\text{con}} = 60Vw(h_i - h_r) = \text{Losses} \quad (8-1)$$

V = cfm air reduced to 70° F and 29.92 in mercury pressure;

w = weight of 1 cu ft of air at 70° and 29.92 in. mercury pressure and of some relative humidity, pounds per cubic foot;

h_i = enthalpy associated with 1 lb of inlet air, Btu per pound;

h_r = enthalpy associated with 1 lb of room air, Btu per pound.

If it is not advisable to use this exact method the air may be considered as free from moisture, and with V as the air at 70° F and standard pressure the expression reduces to

$$q_{\text{con}} = \frac{60V}{55.6} (t_i - t_r) = \text{Losses} \quad (8-2)$$

This heat is equated to the losses from the room, including that for leakage air, as it has been found by Rowley that almost the full air

leakage will occur even into room under pressure in the plenum system.

In this equation either of the quantities V or t_i may be fixed by the designer and the other quantity found. If V is the volume of fresh air brought in, fixed by the occupancy according to laws or ordinances, the equation is used for t_i . If the resulting t_i be too great, V must be increased by recirculated air to give a lower value to t_i . If t_i is fixed at some desired value, the equation gives the volume which must be introduced. When the fresh air is fixed by the A.S.H.V.E. code (10 cfm per person) recirculated air is added to give the volume required by the desired t_i . If the maximum amount of air which can be circulated is less than that required to give this temperature, then the heat loss can be supplied without exceeding t_i only by the use of some direct radiation. This latter is really a modification of the split system as mentioned in Chapter I, in which the temperature of the entering air is that of the room and all the heat loss is cared for by direct radiation.

In general the volume V would be fixed and then t_i is determined from the equation.

The heat supplied by the heating unit, however, is not only that given above but the heating surface must add also the heat necessary to bring the air from the supply temperature to the room temperature. Reducing the volume of outside air and that of the recirculated air to the volume each would occupy at 70° and standard pressure, the following gives the heat supplied by the heater.

Radiator heat per hr

$$\begin{aligned}
 &= 60w[V(t_i - t_r) + V_o(t_r - t_o) + V_{\text{rec}}(t_r - t_i')]0.24 \\
 &= \text{Heat losses} + 60w[V_o(t_r - t_o) + V_{\text{rec}}(t_r - t_i')]0.24 \\
 &= \frac{60}{55.6} [V(t_i - t_r) + V_o(t_r - t_o) + V_{\text{rec}}(t_r - t_i')] \quad (8-3)
 \end{aligned}$$

The subscripts i , r , o and i' refer respectively to inlet, to the room, to outside and to recirculation inlet.

This heat from the radiator is given in Btu per hour, and by using it in this form or reducing it to equivalent pounds of condensate or to E.D.R., the size of the indirect pin radiator, aerofin heater, vento heater or extended surface radiator may be selected, the depth of the unit or number of parts in series for air passage being fixed by the temperature desired. In an indirect system which has a source of heat at one location and in which the temperature t_i for different rooms is not the same, as has been explained earlier, the heater is made up of two portions, one, a tempering coil, and the other, a heating coil. By mixing varying

quantities of air at the temperatures resulting from these coils in the single- or double-duct system, any desired t_i may be obtained. Another arrangement is to have a single duct supplying air at a temperature of the lowest t_i and then using an aerofin heater at the base of each flue to give the desired higher value of t_i for the room into which it discharges. In the gravity system each room has its own temperature of inlet.

The air from the heaters discussed in Chapter IV is carried through ducts to the various rooms as shown in Figs. 2, 3, 4, 5, 10 and 14. The ducts must be arranged so as to reduce the friction to a minimum. All changes in sections must be gradual, and when branches are taken off, the section of the pipe must be provided with a splitter or change in area so that the branch will receive its proper supply. Elbows must be of such a form that friction is made small, and where dampers are introduced they should be formed to control quantity with as little friction as possible and without the production of noise.

These arrangements are indicated in Fig. 143. The figure illustrates a heater requiring a large frontal area which necessitates a gradually diverging cone in front of it and a rather sharp converging cone leading into the fan.

From the fan a diverging cone gradually reduces the velocity of discharge by an increase in section. Where air is taken from the trunk or main duct, the area of the duct at this point or just beyond must be reduced to care for the smaller quantity of air. The radius R of the center line of the elbows is shown as well as the width W measured in the direction of a radius. When the ratio of R to W for an elbow is small, friction is large, so that vanes or splitters are introduced into the elbow to increase this ratio for the

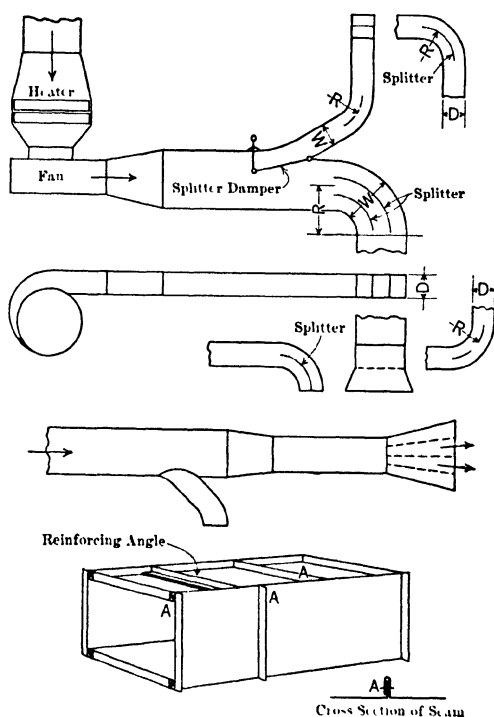


FIG. 143.—Air Piping Details.

separable parts of the air stream. The splitter is not needed when R/W is 0.8 or greater. The ducts are made with flanges as shown in the figure, and those of adjoining sections are lapped over each other in the manner shown.

The resistance offered by the ducts to the flow must be overcome by the draft from the heated column of air in the gravity system where low velocities may be sufficient for proper performance, or by fans in addition to this when space requirements necessitate smaller ducts and higher velocities.

The head produced by the chimney effect from the hot air is similar to that from hot water, and the literal equations for each are the same for L feet of column height.

$$h \text{ in ft of air} = \frac{\text{Pressure per sq ft}}{\frac{w_R + w_h}{2}} = 2L \frac{(w_R - w_h)}{w_R + w_h} \quad (8-3)$$

$$h \text{ in in. of water at } 70^\circ \text{ F per ft of column} = 12 \frac{w_R - w_h}{62.3} = \frac{w_R - w_h}{5.19} \quad (8-4)$$

Since air is a perfect gas

$$w = \frac{p}{53.34T} = \frac{14.7 \times 144}{53.34T}$$

and

$$h = \frac{14.7 \times 144}{53.34 \times 5.19} \left[\frac{1}{T_R} - \frac{1}{T_h} \right] = 7.65 \left[\frac{1}{T_R} - \frac{1}{T_h} \right] \quad (8-5)$$

h = inches of water at 70° F per foot of column;

T_R = absolute temperature of rooms, F° degrees;

T_h = absolute temperature of hot air, F° degrees;

w_R = weight of 1 cu ft of room air;

w_h = weight of 1 cu ft of hot air.

TABLE 8-1

GRAVITY CIRCULATION HEAD FOR BUILDINGS AT 70° F IN INCHES OF 70° F WATER FOR 10 FT COLUMNS

Temperature t_h Degrees F°	100°	110°	120°	130°	140°	150°	160°	170°	180°
h in.	0.008	0.010	0.013	0.015	0.017	0.019	0.021	0.023	0.025

The loss of pressure in pipes due to friction is measured by gages, Fig. 144, attached at two places on the same pipe, L feet apart. Pressure is measured by a U-tube manometer in which water is often used as the fluid. The tube connection must be absolutely flush at the surface of the duct, and the inlet opening from the duct should be very small. This static pressure is reported as h_i inches of water measured by the difference in levels between the menisci in the legs. Although the pressure is read in inches of water, it may be expressed in feet of air or in ounces per square inch. The relations of these are as follows:

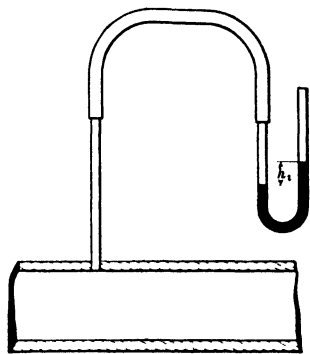


FIG. 144.—U-Tube Manometer.

If h_i represents inches of water,

$$\frac{h_i \times 62.3 \times 16}{12 \times 144} = \frac{h_i}{1.73} = h_o \text{ oz per sq in.}$$

$$1 \text{ oz per sq in.} = 1.73 \text{ in. water at } 70^\circ \text{ F}$$

$$h_i = 1.73h_o \quad (8-6)$$

$$h_o = 0.577h_i \quad (8-7)$$

If this h_i is changed to h feet of air

$$\frac{h_i \times 62.3}{12w} = 5.19 \frac{h_i}{w} = h \text{ ft. of air of density } w \quad (8-8)$$

For air at 70° F and 29.92 in. pressure, $w = 0.07495$

$$h = 69.3h_i \quad (8-9)$$

$$h = 69.3 \times 1.73h_o = 119.9h_o \quad (8-10)$$

If the density w is expressed in terms of the density of 70° F air at 29.92 in., the number expressing this is called relative density, x , and for air of such density the expressions for feet of air become

$$h = \frac{69.3h_i}{x} \quad (8-11)$$

$$h = \frac{119.9h_o}{x} \quad (8-12)$$

The pressure at the wall of the duct, Fig. 145, is measured by the U-tube attached to a pipe A placed perpendicular to the surface of the

duct. Such a pipe as *C* would create eddies and give improper readings. If it is desired to measure the static pressure at various points across the cross-section of the duct, the exploring tube *B* should be connected with a disc normal to the tube and finished with a sharp edge. If the tube *D* with a small opening at the end of a right-angle bend or leg in

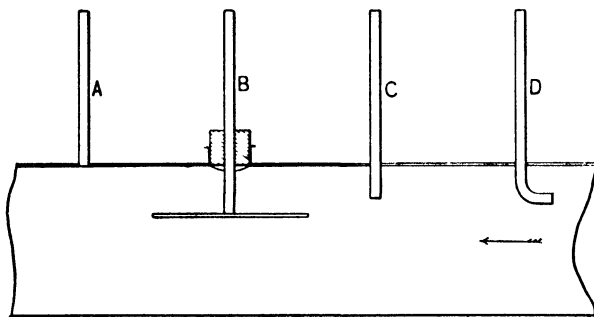


FIG. 145.—Piezometer Connections.

the direction of flow is introduced into the duct and a gage is attached to the outer end, it not only responds to the static pressure, but the gage fluid would rise an additional amount equal to the head produced by the impact of the flowing air. This additional amount is given by

$$h_v = \frac{V^2}{2g} \quad (8-13)$$

This is called the velocity head or velocity pressure, and the sum of the static pressure head and the velocity head shown by the bent tube is called the dynamic head or pressure. By connecting the end of this bent member, Pitot tube, with one end of a U-shaped manometer and the static tube with the other, Fig. 146, the manometer will show the difference between the dynamic pressure and the static pressure, giving the velocity pressure. The device formed of these two tubes combined into a single instrument is called a Pitot tube. The four static pressure openings of 0.02 in. diameter on the surface of the outer tube are at such a point of the leg that eddies will not disturb the proper action of the instrument.

In the equation above for velocity head, the head is expressed in feet of fluid flowing and the velocity in feet per second, but the instrument gives pressure measured in inches of water. For air of density w or of relative density x and for velocity in feet per minute the relation in terms of inches of water becomes

$$V = 60\sqrt{\frac{2g \times 5.19h_i}{w}} = 1096\sqrt{h_i/w} \quad (8-14)$$

$$V = 60\sqrt{\frac{2g \times 69.3h_i}{x}} = 4005\sqrt{h_i/x} \quad (8-15)$$

The difference in pressure from that of standard atmosphere is so slight that these formulae may be used in general, but when the instru-

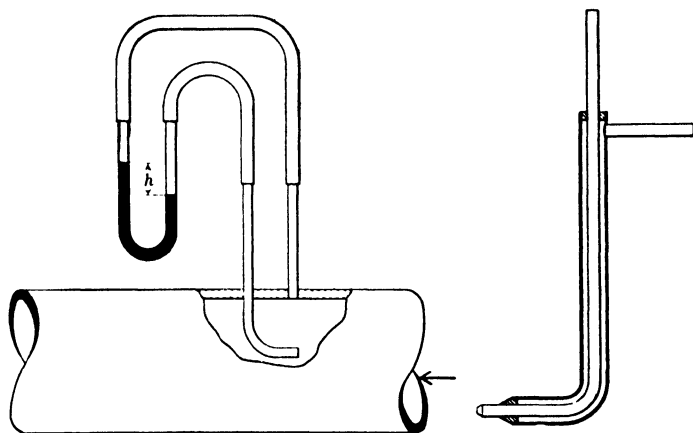


FIG. 146.—Pitot Tubes.

ment is used at altitudes of the city of Denver, the constant must be modified to allow for the atmospheric pressure of much lower value occurring at such places.

Since air is such a light substance, the readings, h_i , are so small for velocities used in practice (0.09 in. of water for 1200 ft per min) that special forms of gages must be used with Pitot tubes or other devices measuring velocities in heating and ventilating systems.

The Wahlen gage, Fig. 147, developed at the Engineering Experiment Station of the University of Illinois,* is very effective in measuring these small differences of pressure. It consists of two bulbs *A* and *B*, an upper U-tube of one of them being connected with the bottom of the other by bent tubes with couplings of rubber tubing. The bulbs are filled with colored alcohol of specific gravity 0.8122, and the inverted U-tube *C* of bulb *A* is filled with kerosene of specific gravity 0.8043. The bulb *A* is mounted on the base of the instrument while the frame which carries the bulb *B* is moved under the control of a micrometer *D*

* See Bulletin 120, Engineering Experiment Station, University of Illinois.

so that the movement of the bulb can be measured to 1/10,000 in. as it is raised or lowered to bring the line of separation between the alcohol from *B* and the kerosene in *C* to a hair-line mark on the reduced portion of the U-tube. Thus, if the Pitot pressure causes the level in bulb *A* to fall, forcing the kerosene below the hair line, the bulb *B* is raised, thus building up pressure and forcing the kerosene back to the hair line. When this line of division reaches the hair line the micrometer reading

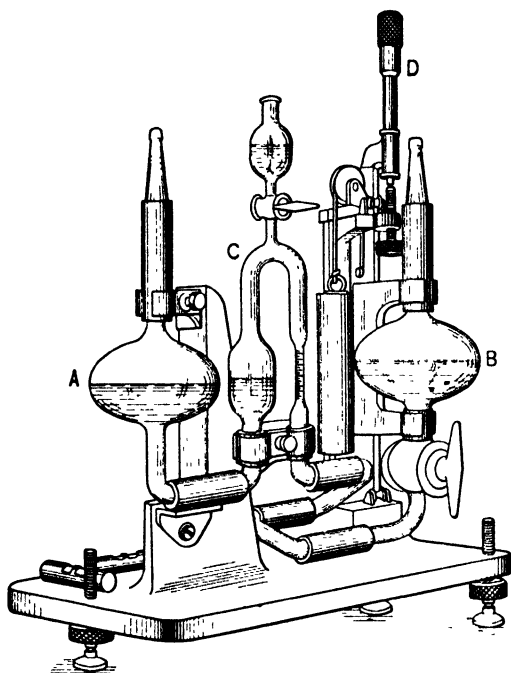


FIG. 147.—Wahlen Gage.

gives the head in inches of alcohol to 0.0001 in. This is the equivalent of 0.00008 in. of water. Of course the zero reading of the gage is taken when both bulbs are open to atmospheric pressure, and this zero reading is taken at the beginning and end of all experiments.

Gages are sometimes built with one inclined tube on the U-tube, so that a small difference in pressure will cause a long movement in the inclined member. The Ellison gage, Fig. 152, is such a gage, in which a vertical reservoir of large size is connected with an inclined tube of small bore, the

density of the liquid and the diameters of the tube and reservoir being so related that readings may be made on the inclined scale to 1/1000 in. of water.

If the velocity head from the Pitot tube is then measured by one of these instruments, it may be converted into velocity in feet per minute by the equation

$$V = 1096\sqrt{h_i/w} \quad (8-14)$$

$$= 4005\sqrt{h_i/x} \quad (8-15)$$

For standard barometric pressure and 70° F air (of relative density unity) the equations give the following velocity heads.

TABLE 8-2

VELOCITY HEAD FROM AIR AT 70° F AND 29.92 IN. PRESSURE IN INCHES OF WATER (AT 70° F) $\times 10^4$

Velocity, feet per minute	100	200	300	400	600	800	1000	1200	1400	1600
Velocity head	6 22	24 9	56 0	99 5	224	398	622	896	1218	1590

From the measured loss in head, studies showed that the formula for the friction loss in pipes carrying air was that used for water.

$$h = 4f \frac{L}{d} \frac{V^2}{2g} = f \frac{L}{\text{Hydraulic radius}} \frac{V^2}{2g} \quad (8-16)$$

h = head lost in L feet or friction, feet of air flowing;

d = diameter in feet = $4 \times$ hydraulic radius;

Hydraulic radius = inside area of flue or pipe divided by inner perimeter, feet;

g = acceleration of gravity, feet per second²;

f = friction factor, a non-dimensional number depending on Reynolds number;

V = velocity, feet per second.

In all problems relating to air flow it must be realized that a modification of the Bernoulli equation for fluid flow holds. That is

Energy per pound of fluid at point 1 + Gains - Losses = energy per pound at point 2.

The energy of 1 lb of air at point 1 consists of three elements:

(a) Kinetic energy = $\frac{V_1^2}{2g}$ in foot-pounds per pound (velocity head);

(b) Internal energy = u_1 in foot-pounds per pound;

(c) Elevation energy = e_1 foot-pound per pound.

The work done on 1 lb of air by the static pressure p_{s1} behind the air at entrance and against the final static pressure p_{s2} at the second point is

$$p_{s1}v_1 - p_{s2}v_2 = h_{s1} - h_{s2} \text{ in ft-lb per lb}$$

where h_{s1} equals the static head in feet of air which is foot-pounds per pound and v is the specific volume, cu ft per pound. The friction loss is

$$\frac{4fLV_m^2}{d2g} \text{ in ft-lb per lb}$$

hence

$$\frac{V_1^2}{2g} + u_1 + e_1 + p_{s1}v_1 - p_{s2}v_2 - 4f \frac{L}{d} \frac{V_m^2}{2g} = \frac{V_2^2}{2g} + u_2 + e_2$$

If heat is added, 778.6 q_s ft-lb per lb, or if work of W foot-pounds per pound is added from a fan these would be added to the left-hand side of the equation.

Now since $u_1 + p_{s1}v_1 = 778.6h_1$ (enthalpy per pound) the equation becomes

$$\begin{aligned} \frac{V_1^2}{2g} + 778.6h_1 + e_1 - 4f \frac{L}{d} \frac{V_m^2}{2g} + \text{Heat or work added} \\ = \frac{V_2^2}{2g} + 778.6h_2 + e_2 \quad (8-17) \end{aligned}$$

This equation is the basis of most of the theory of air flow. The equation may be simplified if there is little change in temperature, if no heat and work are added and if the pipe is level. In this case

$$u_1 = u_2, \text{ giving } 778.6(h_1 - h_2) = h_{s1} - h_{s2} \quad \text{and} \quad e_1 = e_2$$

The equation becomes

$$\frac{V_1^2}{2g} + h_{s1} - 4f \frac{L}{d} \frac{V_m^2}{2g} = \frac{V_2^2}{2g} + h_{s2} \quad (8-18)$$

That is, the dynamic pressure or dynamic head at point 1 minus the friction head loss (or any other loss) between 1 and 2 is equal to the dynamic head at 2. If the friction is very small then the dynamic pressures at two points are of the same value and velocity pressure or static pressure can be changed partially to the other form at some point 2 of the system by changing the area of the pipe, the sum of these pressures or the dynamic pressure remaining unaltered if friction is neglected. The difference between these dynamic heads at points 1 and 2 gives the loss in the device considered between the points. Dynamic pressure is sometimes called total pressure.

In computing the loss of head due to friction in ducts of smooth metal surfaces the value of f is obtained from the Reynolds number

$$\left(\frac{Vdw}{\mu} \quad \text{or} \quad \frac{4V \text{ Hydraulic radius } w}{\mu} \right)$$

by the same curves as those used in Chapter VII for water.

When the Reynolds number (R.N.) is less than 2000 the air flows in stream lines and the loss in friction varies directly as the velocity, but when this value is exceeded turbulent flow occurs and the friction varies as the square of the velocity. The value of the friction factor f and its law of variation are very different in these two cases but Eq. (8-16) holds and the value of f for the formula is found for the R.N. of the flue with any velocity.

An investigation of the effect of temperature change from 70° F to 160° F at standard atmospheric pressure showed that for a velocity of 150 fpm in pipes of different diameters the value of f for each of the diameters increased about 6½% with this increase of temperature, while at 1200 fpm the increase was 5½%. The variation in values of f due to the extreme changes in either velocity or diameter may amount to decreases of 35% with the increases of these quantities, and with simultaneous changes of both of them the decrease may amount to 50%. For this reason the friction drop in pipes of different diameters has been computed in inches of water at 70° F per 100 ft of length for air at 70° F and at a pressure of 29.92 in. of mercury moving at different velocities. Eq. (8-16) for this temperature and pressure and for velocities in feet per minute becomes

$$h_i \text{ in inches of water in 100 ft} = \frac{1}{3350} f \frac{V^2}{d} \quad (8-19)$$

V = velocity in feet per minute;

d = inside diameter of circular pipe in inches;

f = friction factor dependent on Reynolds number.

The relation between cubic feet per minute, velocity and diameter is given by

$$V = \frac{144 \text{ cfm}}{\pi \frac{d^2}{4}} = \frac{183 \text{ cfm}}{d^2} \quad (8-20)$$

hence

$$h_i \text{ in inches of water at 70° in 100 ft} = 10f \frac{(\text{cfm})^2}{d^5} \quad (8-21)$$

These results of Eqs. (8-20) and (8-21) are plotted logarithmically in Fig. 148, which gives the relations between cubic feet per minute and friction head as well as velocity, for pipes of different diameters, and may be used in the selection of pipe sizes for desired velocities or friction losses.

When the ducts are of masonry the tabular values of loss are multiplied by 1.125 on account of the greater friction.

As the value of $4f$ is sometimes taken as 0.02 (a poor approximation) the head becomes

$$h = 0.02 \frac{L V^2}{d 2g} = \frac{L V^2}{50d 2g} \quad (8-22)$$

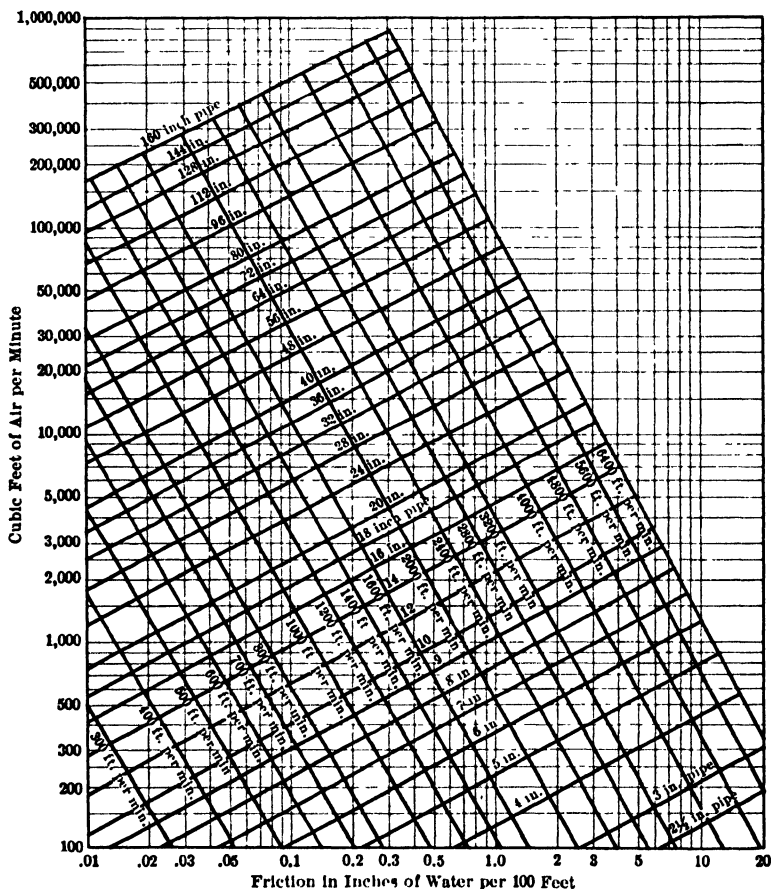


FIG. 148.—Friction in 100 ft of Pipes Carrying Air, Capacity and Velocity.

This means that the velocity head $\frac{V^2}{2g}$ is the friction loss in 50 diameters of length in round pipes approximately.

For cfm of air at temperatures and pressures other than 70° and 29.92 in. the friction head h , selected from the table for this cfm must

be multiplied by the relative density x to get the loss for such air due to friction expressed in inches of 70° water. If the air of relative density x is reduced to the volume it would occupy at 70° and the loss h_i is found from the chart for this new volume, then the h_i for the actual air will be that from the chart divided by x .

For rectangular pipe, the value $4/d$ of the expression for friction loss is replaced by $\frac{1}{\text{Hydraulic radius}}$

$$\text{Hydraulic radius} = \frac{\text{Area}}{\text{Perimeter}} = \frac{l_s l_l}{2(l_s + l_l)} \quad (8-23)$$

l_l = length of long side, inches;

l_s = length of short side, inches.

This gives the loss in 100 ft of pipe expressed in inches of water at 70° as

$$h = 3.09f' \frac{l_s + l_l}{(l_s l_l)^3} (\text{cfm})^2 \quad (8-24)$$

If the friction drop in a rectangular pipe is the same as that in a circular pipe when carrying the same quantity of air Eqs. (8-21) and (8-24) may be equated

$$\begin{aligned} 3.09f' \frac{(l_s + l_l)}{(l_s l_l)^3} &= 10 \frac{f}{d^5} \\ 3.24 \frac{f}{f'} \frac{(l_s l_l)^3}{l_s + l_l} &= d^5 \end{aligned} \quad (8-25)$$

The friction factor f depends on the Reynolds number

$$\frac{dVw}{\mu} \quad \text{or} \quad \frac{2l_s l_l V'w}{(l_s + l_l)\mu}$$

and the ratio of these would be

$$\frac{dV}{2V'} \frac{(l_s + l_l)}{l_s l_l} \quad \text{or} \quad \frac{2(l_s + l_l)}{\pi d}$$

Hence, the two friction factors, f and f' , are not the same. In general the hydraulic radius of the equivalent rectangular duct and the velocity in it are less than those for the circular pipe. The Reynolds number for the rectangular pipe for a ratio of sides 1 to 1 is about 0.86 of the Reynolds number for the equivalent circular pipe, while for a ratio of the sides of the rectangle of 1 to 4, this is 0.66, and for 1 to 8 it is 0.48. These variations of the Reynolds number will produce ratios

of f'/f of 1.04, 1.115 and 1.205 respectively. Using these ratios the following result:

$$\text{For } l_i = l_s; d^5 = 3.24 \times \frac{1}{1.04} \times \frac{1}{2} l_s^5; d = 1.092 l_s$$

$$\text{For } l_i = 4 l_s; d^5 = 3.24 \times \frac{1}{1.115} 12.8 l_s^5; d = 2.065 l_s$$

$$\text{For } l_i = 8 l_s; d = 3.24 \times \frac{1}{1.205} \times 56.9 l_s^5; d = 2.74 l_s$$

For a value of l_s of 3, the values of d become 3.3, 6.2 and 8.2.

For rapid determinations of d Table A-18 has been computed, giving the diameters of circular ducts having the same capacity as rectangular ducts with the same friction drop in a given length.

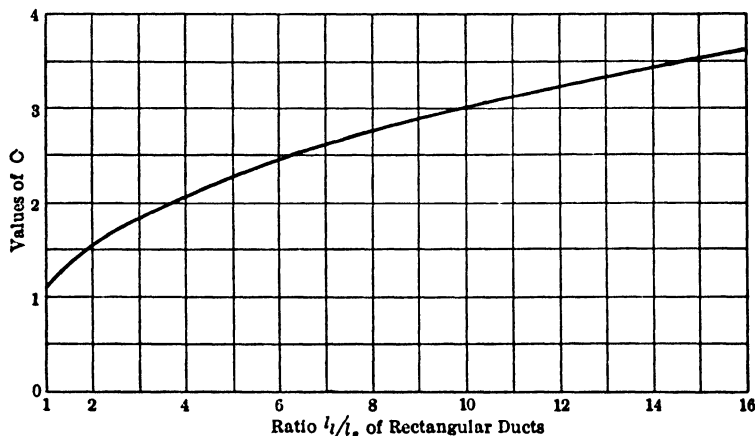


FIG. 149.—Coefficient for Round Pipe Equivalent to Rectangular Duct.

The diameters of any size ducts having ratios in Table A-18 can be computed by proportion or by use of Fig. 149, which gives the multiplier C of the short side of the rectangle of different ratios of l_i/l_s by which the diameter of the equivalent circular pipe is determined

$$d = C l_s \quad (8-26)$$

If f be considered as constant, the formula for friction loss in round pipes is

$$h_1 = \frac{CL(\text{cfm})^2}{d^5} \quad (8-27)$$

From this it is seen that friction drop varies directly as the length and the square of the discharge and inversely as the fifth power of the

diameter. For a given drop in a length L the diameter will vary as $2/5$ power of the discharge. This suggests that if a duct has been designed to care for $(\text{cfm})_1$, with a drop h_1 in a length L_1 , another duct of the same proportions with a drop in pressure of h_2 in a length L_2 when carrying $(\text{cfm})_2$ will have its dimensions determined by multiplying each side of the original cross-section by

$$\left(\frac{\text{cfm}_2}{\text{cfm}_1}\right)^{5/2} \left(\frac{L_2}{L_1}\right)^{1/2} \left(\frac{h_1}{h_2}\right)^{1/2}$$

The velocities to be used in practice for offices, hotels, schools and public buildings are given in Table 8-3.

TABLE 8-3
VELOCITIES IN PARTS OF AIR SYSTEM

Vertical grilles:

On gross area.....	200 to 400 fpm
On net area.....	300 to 550 fpm

Horizontal grilles:

On gross area.	130 to 175 fpm
On net area.....	200 to 250 fpm
Flues	500 to 750 fpm
Ducts.....	800 to 1200 fpm

Velocities have been designated by Carpenter as suggested in Loomis' Meteorology,

Just perceptible.....	175 fpm
Quietly pleasant.....	230 fpm
Pleasant—brisk.....	1100 fpm
Very brisk.....	2200 fpm
High wind.....	3100 fpm
Very high wind....	4000 fpm

Where noise is not objectionable, as in industrial installations, these velocities may be increased to 2 times the values above, for stores to $1\frac{1}{2}$ times the values. The areas of the exhaust ducts are made 120% of the areas of the inlet ducts which they serve.

In determining the loss of head in duct systems the losses due to entrances, heating coils, air washers, cone-shaped connecting sections, elbows, branch connections, splitters and grilles must be known. The losses due to these have been determined by a number of experimenters and reported in equivalent lengths of pipe or as percentages of the velocity head, $\frac{V^2}{2g}$. The latter method is preferred, and the following coefficients C are used in the formula

$$\begin{aligned}
 h_i &= C \left(\frac{V}{4005} \right)^2 \text{ for } 70^\circ \text{ air at } 29.92 \text{ in.} \\
 &= xC \left(\frac{V \text{ actual}}{4005} \right)^2 \\
 &= \frac{C}{x} \left(\frac{V \text{ at } 70^\circ \text{ F}}{4005} \right)^2
 \end{aligned}
 \tag{8-27}$$

V = velocity in feet per minute;

x = relative density = ratio of density of air at temperature t to density at 70° and 29.92 in.

h_i = inches of water at 70° .

TABLE 8-4
FRICTION LOSSES IN AIR DUCT FITTINGS

VALUES OF C FOR LOSS = $\frac{CV^2}{2g}$

Loss at entrance (0.1 to 0.5) . . .	0.50	Branches (135°)	0.15
Grilles (50% free area)	1.50	Washers 16 to 24	20.0
(67% free area)	0.75	Converging cones	0.04
		(on velocity in small end)	

ELBOWS*

(W is width in direction of radius R to center line.)

Circular		Square	
R/W	C	R/W	C
0.0	0.87	0.0	1.25
0.5	0.76	0.5	0.95
0.8	0.34	0.8	0.28
1.0	0.26	1.0	0.18
1.5	0.17	1.5	0.09
2.0	0.15	2.0	0.08
3.0	0.14	3.0	0.07

* Fan Engineering, Buffalo Forge Company, after F. L. Busey.

For rectangular ducts when W is less than the depth perpendicular to W the loss is slightly less than that given for square ducts, but when W is greater than the depth, the value of C is so great that the elbow is divided up into a number of passages by splitters to make the separate

sections square or rectangular with W greater than D . The splitters must be arranged so that the value of R/W for each section is the same.

Thus for a single splitter dividing W into w_1 and w_2 for an inner radius of R_1

$$\frac{R_1 + \frac{w_1}{2}}{w_1} = \frac{R_1 + w_1 + \frac{w_2}{2}}{w_2}$$

$$R_1 w_2 + \frac{w_1 w_2}{2} = R_1 w_1 + w_1^2 + \frac{w_1 w_2}{2}$$

$$w_2 = \frac{w_1^2 + R_1 w_1}{R_1}$$

but

$$w_2 = W - w_1$$

Hence

$$WR_1 = w_1^2 + 2R_1 w_1$$

$$R_1^2 \left[\frac{W}{R_1} + 1 \right] = (w_1 + R_1)^2 = (\text{radius of splitter, } \rho)^2$$

$$\frac{\rho_1}{R} = \left[\frac{W}{R_1} + 1 \right]^{1/2} \quad (8-28)$$

For two splitters a third term is added

$$\frac{R_1 + w_1 + w_2 + \frac{w_3}{2}}{w_3} = \frac{R_1 + \frac{w_1}{2} + \frac{w_2}{2} + \frac{W}{2}}{W - w_1 - w_2}$$

since

$$w_3 = W - w_1 - w_2$$

The first and third give

$$R_1 W - R_1 w_1 - R_1 w_2 + \frac{W w_1}{2} - \frac{w_1^2}{2} - \frac{w_1 w_2}{2}$$

$$= R_1 w_1 + \frac{w_1^2}{2} + \frac{w_1 w_2}{2} + \frac{W w_1}{2}$$

$$R_1 W = w_1^2 + w_1 w_2 + 2R_1 w_1 + R_1 w_2$$

Substituting the value of w_2 which still holds

$$R_1^3 \left[\frac{W}{R_1} + 1 \right] = w_1^3 + 3R_1 w_1^2 + 3R_1^2 w_1 + R_1^3 = (w_1 + R_1)^3$$

$$\frac{\rho_1^3}{R_1} = \left[\frac{W}{R_1} + 1 \right]^{3/2} \quad (8-29)$$

Working from the outer side the radius of the second splitter becomes

$$\frac{\rho''_2}{R_1} = \left[\frac{W}{R_1} + 1 \right]^{36} \quad (8-30)$$

The general formula* becomes

$$\frac{\rho \text{ for } x, u}{R_1} = \left[\frac{W}{R_1} + 1 \right]^{\frac{x}{u+1}} \quad (8-31)$$

where x is the particular splitter of u splitters. The purpose of these is to reduce the value of W/R and so give a smaller value of C .

There is not much advantage on making R for elbows much greater than $1.5W$. When the value of R must be made very small the width of the discharge end of the elbow is reduced to 75% of the width at entrance and then the flue is gradually enlarged. This is called a Venturi elbow in Fan Engineering.

By the Bernoulli equation the loss due to sudden enlargement is given as

$$\frac{1}{3600} \left(\frac{V_1 - V_2}{2g} \right)^2 \text{ in feet of air}$$

or

$$\left(\frac{V_1 - V_2}{4005} \right)^2 \text{ in inches of } 70^\circ \text{ water} \quad (8-32)$$

From the use of the Bernoulli equation, the loss due to sudden contraction as that which occurs when a small pipe is attached to a flat plate on the end of a large pipe is

$$h_i = \left(\frac{V}{4005} \right)^2 \left(\frac{1}{C_1^2} - 1 \right) \quad (8-33)$$

The value of the $\left(\frac{1}{C_1^2} - 1 \right) = C$, which depends on the ratio of the area of the small pipe A_s to that of the large pipe A_l , is given in Fan Engineering as 0.20 for A_s/A_l of 0.7 to 0.48 for A_s/A_l of 0.1. As A_s/A_l approaches 0 the value of C approaches 0.5, the loss at entrance from a plenum chamber into a pipe with a sharp edge. With a rounded entrance this loss becomes small, C may be taken as 0.1.

Converging Pipe Section. The loss due to impact in a converging section is very small, and may be taken as 3 or 4% of the velocity head

* Fan Engineering, Buffalo Forge Company.

in the smaller section, the friction loss being taken for its length at the velocity in the smaller pipe.

Diverging Pipe Sections. In the design of air ducts, velocities are decreased by means of diverging sections for which the Bernoulli equation with constant temperature is

$$h_s + \frac{V_s^2}{2g} - h_l - \frac{V_l^2}{2g} = \text{loss}$$

$$\frac{V_s^2 - V_l^2}{2g} = h_l - h_s + \text{loss} \quad (8-34)$$

Since the velocities are known from the areas as there is little change in specific volume, the change in velocity head should equal the increase of pressure head were there no losses. The complete losses are given in Fan Engineering as a percentage of the change in kinetic energy or velocity head. This loss is given for the most efficient diverging connections where the length L bears various relations to the smaller diameter, D .

TABLE 8-5
DIFFUSER DATA

L/D	Best A_l/A_s	% of Change Lost
$\frac{1}{2}$	1.45	73
1	1.70	62
$1\frac{1}{2}$	1.90	55
2	2.05	50
3	2.30	43
4	2.50	37
5	2.65	34

The ducts and fittings are usually made of galvanized sheet iron, the thickness or gage of the sheets varying with size. The Buffalo Forge Company recommends the thicknesses given in Table A-19. See Table A-23 for gages.

The largest of these ducts should be braced with angle iron or by angle seams, Fig. 143. If the joints of circular ducts are slip joints they should be arranged with an outer overhang of slightly larger diameter, although for rectangular ducts seams are made of the standing form. The ducts are firmly supported to prevent noise as well as to eliminate sagging. All construction joints of sections and those between must be made tight.

The register faces or grilles, Fig. 150, are made of cast or pressed metal. Various sizes and designs are available. They are constructed

with a frame which fits into the air duct. The net open area varies with the design. In general the net or open area is about 66% of the duct area.

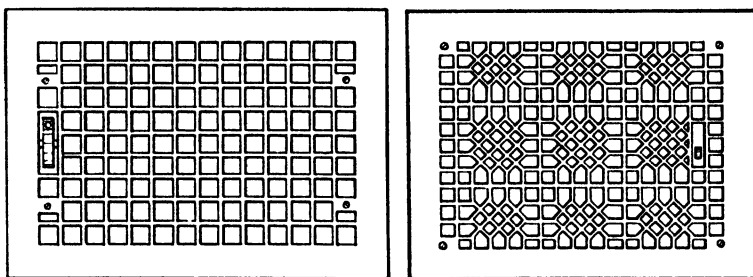


FIG. 150.—Grilles or Register Faces.

Measurement of Air. Pitot Tubes. The quantity of air flowing in any pipe may be measured by using the Pitot tube to determine the velocity at various radii on two lines at right angles.

$$V_x = 4005 \sqrt{\frac{h_i}{x}}$$

In this h_i is the average gage reading at the same radius on the two diameters. This velocity applies to an element of cross-sectional area, $2\pi r_x dr$, so that

$$\text{Volume per minute} = \int 2\pi r_x V_x dr = \pi \int V_x d(r_x^2) \quad (8-35)$$

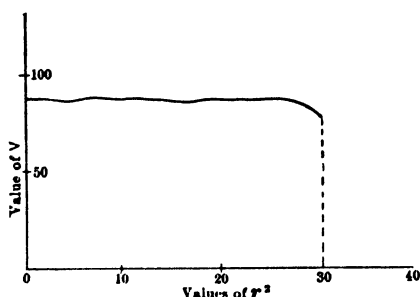


FIG. 151. — Plot of V vs r^2 from Pitot Readings.

This means that, if the velocity be plotted against r^2 , the area below this curve, Fig. 151, if multiplied by π will be the volume per minute. If it is desired to take the average of all Pitot readings as the average velocity of the whole pipe, the tube should be placed at the center of rings of equal area. Thus suppose that 3 Pitot positions were to be desired on each radius; then the radii to these tube positions and the boundary radii would be

$$r_1 = r\sqrt{\frac{1}{8}} = 0.408r$$

$$r_2 = r\sqrt{\frac{3}{8}} = 0.707r$$

$$r_3 = r\sqrt{\frac{5}{8}} = 0.912r$$

$$r_{1b} = r\sqrt{\frac{1}{3}} = 0.578r$$

$$r_{2b} = r\sqrt{\frac{2}{3}} = 0.817r$$

$$r_{3b} = r = 1.000r$$

Orifices, of the forms shown in Fig. 152, are used to measure the volume of air delivered from a pipe. For these the Bernoulli equation (8-17) with no friction, heat, work or change in elevation reduces to

$$\frac{V_2^2 - V_1^2}{2g} = 778.6(h_1 - h_2)$$

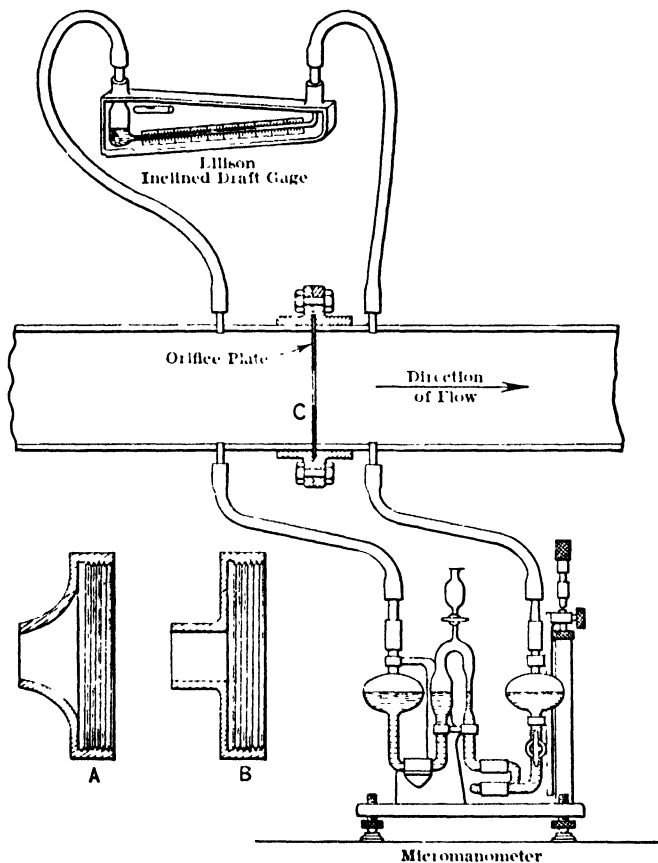


FIG. 152.—Orifices—Ellison and Wahlen Gages.

V = velocity in feet per second;

h_1 = enthalpy in Btu per pound at higher pressure;

h_2 = enthalpy in Btu per pound at lower pressure.

If V_1^2 is negligible compared with V_2^2 , as would hold in Fig. 152, B and C , this becomes

$$V_2 = 223.7\sqrt{h_1 - h_2}$$

and h_1 and h_2 are found at the same entropy value if there is no friction

$$778.6h_1 = \frac{1.4}{0.4} p_1 v_1, 778.6h_2 = 3.5 p_2 v_2, p_1 v_1^{1.4} = p_2 v_2^{1.4}$$

and

$$V_2 = 15 \sqrt{p_1 v_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{0.4}{1.4}} \right]}$$

Whenever $p_2 < 0.53p_1$ the pressure at the lower side of the orifice is $0.53p_1$ and the enthalpy h_2 is found for that pressure and $V_2 = 6.04 \sqrt{p_1 v_1}$. To get the weight of air discharged, the area A of the orifice is multiplied by the velocity V_2 and the density at $0.53p_1$ after this adiabatic expansion.

$$\frac{w_{0.53p_1}}{w_{p_1}} = (0.53)^{\frac{1}{1.4}} = 0.635$$

$$w_{0.53p_1} = 0.635w_{p_1}$$

Thus $0.63AV_2w_{p_1} = W =$ discharge in pounds per second. This may be reduced to Fleigner's equation for air

$$W = \frac{0.53Ap_1}{\sqrt{T_1}} \quad (8-36)$$

When $p_2 > 0.53p_1$ both p_1 and p_2 appear in the second Fleigner equation as

$$W = 1.06A \sqrt{\frac{p_2(p_1 - p_2)}{T_1}} \quad (8-37)$$

$$V = Wv$$

These equations are multiplied by coefficients to care for friction, 0.62 being used for C , Fig. 152, 0.75 to 0.84 for B and 0.98 for A . The gages of Fig. 152, C , indicate the methods of measuring $(p_1 - p_2)$ for use in these formulæ and indicate that, if the orifice is calibrated for a given pressure drop, pressure and temperature curves of W could be plotted for definite temperatures T_1 and final pressure p_2 .

Venturi Meter. When a convergent pipe section of short length, Fig. 153, is followed at the small end or throat by a divergent section of much longer length with sides at 7° , the combination is called a Venturi meter. This is used to determine the quantity of flow of all kinds of fluids by the application of the Bernoulli equation above, measuring the temperature and the static pressure at the large section A_1 , and the dif-

ference in static pressure between that at the point A_1 and that at the throat A_2 , as well as the areas at A_1 and the ratio of areas A_1/A_2 .

This equation is

$$\frac{V_2^2 - V_1^2}{2g} = 778.6 [\text{Enthalpy}_1 - \text{Enthalpy}_2]$$

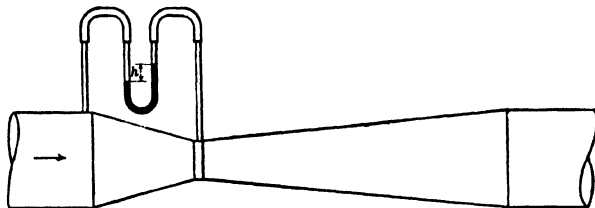


FIG. 153.—Venturi Meter.

This reduces to Eq. (8-38) for weight per second or to Eq. (8-39) for volume per second with adiabatic action on air.

$$\frac{W}{C} = 110 \frac{A_1}{v_1} \sqrt{\frac{T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{0.4}{1.4}} \right]}{\left(\frac{p_1}{p_2} \right)^{\frac{2}{1.4}} \left(\frac{A_1}{A_2} \right)^2 - 1}} \quad (8-38)$$

$$\frac{\text{Volume}}{C} = V_1 A_1 = W v_1 = 110 A_1 \sqrt{\frac{T_1 \left[1 - \left(\frac{p_2}{p_1} \right)^{\frac{0.4}{1.4}} \right]}{\left(\frac{p_1}{p_2} \right)^{\frac{2}{1.4}} \left(\frac{A_1}{A_2} \right)^2 - 1}} \quad (8-39)$$

C = coefficient of discharge = 0.98;

A = area in square feet;

p = pressure in pounds per square foot;

T = absolute temperature in degrees F.

For small differences in pressure Eq. (8-39) reduces to

$$\text{Vol}_1 \text{ per sec} = 8.02 C A_1 \sqrt{\frac{(p_1 - p_2) v_1}{\left(\frac{A_1}{A_2} \right)^2 - 1}}$$

Electric Meter. The volume or weight of air flowing may also be measured by the Thomas electric meter in which the air is passed

through an electric grid and heated, the temperature being measured in advance of the grid and beyond it by resistance thermometers composed of wires crossing the full area. If the watts of energy and the two temperatures are measured, the following results:

$$\text{Volume}_1 w_1 (h_2 - h_1) = \text{Volume}_1 w_1 c_p (t_2 - t_1) = \frac{3.41 \times \text{Average watts}}{60}$$

Volume₁ = cubic feet per minute of air;

w₁ = weight of 1 cu ft in pounds;

h = enthalpy, Btu per pound;

c_p = specific heat at constant pressure, Btu per pound per degree F.

Anemometer. The discharge from grilles and pipe openings may be measured by the use of an anemometer, Fig. 154. This instrument measures velocity and is rarely used above 1500 fpm. The rotation of the vanes on the shaft by the current of air is recorded by throwing in a train of gears at a given instant and then disengaging the train at the

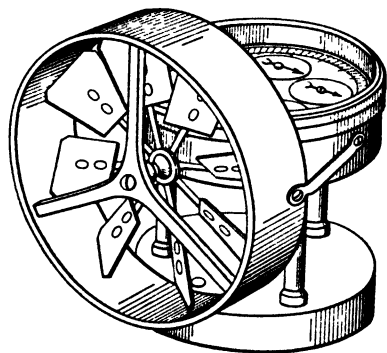


FIG. 154.—Anemometer.

end of a definite time interval fixed by a stop watch. The revolutions per minute or the feet per minute read from the dial of the instrument are found, and a calibration curve gives the correct velocity of the air passing over the instrument in feet per minute or the correction to apply to the reading. In measuring air flow, the area of discharge from a grille or duct end is divided into sections of equal area and the velocity at each of these is determined by placing the anemometer against the

grille or in the opening. The average of these is taken as the average velocity. Some engineers determine the reading of the anemometer during a time interval while the instrument is slowly moved over the face of the grille. This gives the average velocity. Professor L. E. Davies has investigated this method of measuring air and suggests that the average velocity in feet per minute be multiplied by a constant and the product be multiplied by the average of the gross area and the free area of the grille in square feet for outlet grilles, while for exhaust or inlet grilles he used the gross area of the grilles as the anemometer is

affected by all this suction area. He has found that the value of the constants C and K below change slightly with the velocity.*

TABLE 8-6
ANEMOMETER COEFFICIENTS FOR GRILLES (by Davies)

V_1 fpm	150	200	300	400	500	600	700	800
C supply out..	0.952	0.957	0.967	0.977	0.985	0.992	0.998	1.000
K exhaust in. .	0.762	0.772	0.789	0.806	0.820	0.828	0.832	

$$\text{Volume} = CV \frac{A_g + A_n}{2} (\text{supply}) = KVA_g (\text{exhaust})$$

Volume = cubic feet per minute;

V = velocity in feet per minute (corrected);

A_g = gross area of duct in square feet;

A_n = net or free area in square feet.

Velocity by Kata Thermometer. To determine low velocities the kata thermometers of Dr. Leonard Hill may be used, although these were developed to indicate comfort. The thermometer, using alcohol as the fluid, has a large bulb of $\frac{5}{8}$ in. diameter and $\frac{1}{2}$ in. long, with a short stem, 8 in. long, graduated in $\frac{1}{10}$ degrees, from 95° F to 100° F. The bulb is heated in a water bath until the fluid passes the 100° mark and flows into a well at the top, and then after wiping, the bulb is exposed to room air and the time in seconds required for the column to pass from the 100° mark to the 95° mark is noted. This time is a function of the radiation and convection to the surroundings and is a function of the dry-bulb temperature. If a wet kata thermometer be treated in the same way, with a wet finger stall of woven goods covering the bulb, the time taken is dependent upon radiation, convection and evaporation and depends on wet-bulb temperature irrespective of the dry bulb. These times were used to judge comfort.

The rate of temperature change of the dry bulb varies with the velocity of air over the thermometer and by calibration, with reference to room temperature, this time required to reduce the temperature reading from 100° to 95° of the dry-bulb temperature will determine the low velocities of air producing this cooling.

Velocity by Resistance Wire. The voltage applied to a circuit of fixed current having an exposed high-resistance wire from which

* Fan Engineering, Buffalo Forge Co.

heat is being removed by moving air will depend upon the resistance of this coil. This resistance will be less as the velocity of the air increases to remove a greater quantity of heat, cooling the coil and decreasing its resistance. By calibration, the current flowing at fixed resistance, or the resistance at fixed current, could be used to measure velocities.

Gravity System of Indirect Heating. This system, as shown in Fig. 2, uses pin radiators (or their equivalents) placed in sheet-iron housings at the bottoms of the various stacks or groups of adjacent stacks. In applying these stacks the desired temperature of the entering air is assumed, and then the amount of air for the room or for one unit, if several are required for a large room, is computed. The circulating head is next determined by the temperatures and the height between the outlet of the stack and the center of the heater unit.

An equation is then written between the losses in x fittings and pipes and the circulating head.

$$\text{Loss in heater} + \frac{xcV^2}{2g} + xL \text{ (duct loss per ft)} = h_1 \quad (8-40)$$

The values of x , c , L and loss in heater are known or assumed, and the value of V is obtained by trial.

After this first approximation for V , the sizes of the ducts and parts may be found for this value of V after which the more correct values of the coefficients can be found and the equation is again solved for V . If this second value is close to the value first found it will not be necessary to make another solution, otherwise a third approximation or even a fourth must be made.

To employ this method, the gravity system of indirect heating will be applied to the building of Chapter V and the calculation for two rooms will be worked out in detail.

It will be assumed that all air taken in will be fresh air and that air will be delivered at 140° F . In this case the air needed for each room in cubic feet per minute is

$$\text{cfm} = \frac{\text{Heat loss}}{0.018 \times 60 (140 - 70)}$$

The heat given up by radiator = $\text{cfm} \times 0.018 \times 60 \times (140 - 70) = 2 \text{ Heat loss}$

Floor registers are used for the first floor, the head causing flow will be 2 ft and the circulating head will be 0.0034 in. of water. The second-floor rooms will be equipped with wall registers, the head for which is 12 ft and the circulating head 0.020 in. of water.

From the arrangement of the radiator and its connecting casing, Fig. 155, combined with the cellar plan, Fig. 156, it is seen that the resistances to rooms 101 and 205 will be as follows:

Room 101			
Entrance.....	0.10	velocity head	
Velocity.....	1.00	"	"
Screen.....	0.75	"	"
Pipe 12 ft			
Short elbow.....	0.30	"	"
<hr/>			
Total cold.....	2.15		
Pin radiator			
Short elbow.....	0.60	"	"
Pipe 2 ft			
Grille.....	0.75	"	"
<hr/>			
Total hot.....	1.35	"	"
<hr/>			
Final velocity head	1.00		

Room 205			
Entrance.....	0.10	velocity head	
Velocity.....	1.00	"	"
Screen.....	0.75	"	"
Pipe 10 ft			
Short elbow.....	0.30	"	"
<hr/>			
Total cold.....	2.15		
Pin radiator			
Short elbow.....	0.60	"	"
Pipe 11.5 ft			
Short elbow.....	0.60	"	"
Grille.....	0.75	"	"
<hr/>			
Total hot.....	1.95		
<hr/>			
Final velocity head	1.00		

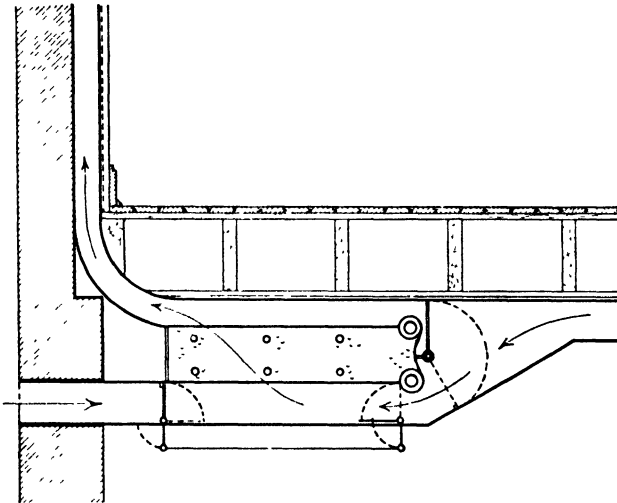


FIG. 155.—Indirect Heater and Casing.

As no velocity is known at this time it will be assumed that the total velocity heads for room 101 will be 2.15 + 1.35 as given + 4.00 velocity heads to care for pipes and heater. The sum of these (7.50) is divided into the 0.0034 inches available

$$\frac{0.0034}{7.5} = 4.5 \times 10^{-4} = \text{Velocity head}$$

As this is at 140° it is reduced to 70° by the relative density 0.883.

$$\text{Velocity head } 70^\circ = 5.10 \times 10^{-4}$$

From table of heads the velocity will be

$$100 \sqrt{\frac{5.10}{6.22}} = 90 \text{ fpm}$$

Using 100 fpm as the velocity as a first approximation, the sizes of the duct and of the heater will be determined so as to find the resistances of these parts.

Heat for room 101—19,740 Btu per hr—2 units each for 9870 Btu per hr.

Heat from 0° to 140° = 2 (Heat loss) = 19,740 Btu

$$\text{cfm} = \frac{19\,740}{60 \times 0.018 \times 140} = 130 \text{ cfm (at } 70^\circ\text{)}$$

$$\text{Area of duct } \frac{130}{1000} = 1.30 \text{ sq ft} = 188 \text{ sq in.} = 8 \text{ in.} \times 24 \text{ in.}$$

Equivalent pipe diameter 14.7 in.

Pipe resistance for pipe 14.7 in. diameter at 100 fpm will be taken at 400 fpm and reduced to 100 fpm as chart does not reach 100 fpm.

$$\begin{aligned}\text{Loss} &= 0.02 \times \left(\frac{130}{100}\right)^5 \\ &= 0.0012 \text{ in. per } 100 \text{ ft}\end{aligned}$$

From an extrapolation on Fig. 88 for $\frac{1}{2}$ in. pins, it is seen that zero air must be allowed to flow over the radiator with $\frac{1}{2}$ in. pins at the rate of 150 cfh per sq ft to give a temperature rise of 140°. This means that the radiator must have an area of

$$\frac{130 \times 60}{150} \text{ or } 52 \text{ sq ft}$$

The heat transfer from the pin radiator with zero degree air at 150 cfm per sq ft is given as 350 Btu on Fig. 88. This gives

$$\frac{19\,700}{350} = 56.3 \text{ sq ft}$$

The mean of these is 54.2 sq ft.

Since the Perfection pin radiators contain 10 sq ft, six of these will be used requiring $36\frac{1}{4}$ in. length \times 18 in. width \times 10 in. The resistance through these at this low velocity will be estimated as 0.001 in. from the table of resistance through Vento heater.

The resistances reduced from 70° air now become:

$$\text{Pipe } 12 \times \frac{0.0012}{100} = 0.00014 \text{ in.} \times 1.152 = 0.00016 \text{ in.}$$

Fittings and initial velocity = 2.15 velocity

head

$$1.152 = 2.47$$

$$\text{Pin radiator} = 0.0010 \text{ in.} \times 1.00 = 0.001$$

$$\text{Pipe } 2 \times \frac{0.0012}{100} = 0.00002 \times 0.88 \text{ in.} = 0.00002$$

$$\text{Fittings } 1.35 \text{ velocity head} \times 0.88 = 1.20$$

$$0.00118 \text{ in.}$$

$$3.67 \text{ velocity head}$$

$$\text{Available head} = 0.0034 \text{ in.}$$

$$\text{Pipe and radiator} = 0.00118$$

$$\text{Head for velocity} = 0.00222 \text{ in.}$$

$$\text{Velocity head} = \frac{0.0022}{3.67} \text{ in.} = 0.00065 \text{ in.}$$

$$\text{Velocity head for 100 ft} = 0.0006 \text{ in. of water.}$$

The velocity required is then

$$\sqrt{\frac{0.00065}{0.0006}} \times 100 = 104 \text{ fpm}$$

This is so close to 100 that if this velocity is used to determine sizes there will be no need of a recomputation of friction.

$$\text{Size of duct} = \frac{130 \times \frac{1}{1.152}}{104} = 1.09 \text{ sq ft} = 156 \text{ sq in. or } 8 \text{ in.} \times 20 \text{ in.}$$

$$\text{Gross area register face} = \frac{130}{0.66} \times \frac{1}{0.883} \times \frac{1}{104} = 2.15 = 309 \text{ sq in. or } 18 \text{ in.} \times 18 \text{ in.}$$

For room 205

Heat required 8740 Btu per hr

Heat from 0° F to 140° F = 2 × 8740 = 17,480 Btu.

$$\text{cfm} = \frac{17480}{60 \times 0.018 \times 140} = 115 \text{ cfm at } 70^\circ$$

Since there is a greater head causing flow the velocity will be taken as 200 fpm, giving the air duct

$$\frac{1}{2} \frac{1}{6} \frac{5}{8} \times 144 = 83.0 \text{ sq in. or } 8 \text{ in.} \times 10\frac{1}{2} \text{ in. for entrance}$$

Because of the 4 in. partitions the flue leading to the second floor will have to be made up of two flues each $3\frac{1}{2} \times 12$. The equivalent pipe for the duct and flues will be 10 in. and 6.7 in., respectively.

The resistances in these pipes at 200 fpm are 7.8×10^{-5} in. and 1.3×10^{-4} in. per ft of length. The resistance in the heater will be 0.001×4 as the clear space remains the same. The resistances reduced from 70° air are

$$\text{Pipe } 10 \times 7.8 \times 10^{-5} = 0.00078 \times 1.152 = 0.0009 \text{ in.}$$

$$\text{Fittings} = 2.15 \times 1.152 = 2.47$$

$$\text{Pin radiator } 0.001 \times \left(\frac{200}{100}\right)^2 = 0.0040 \text{ in.}$$

$$\text{Pipe } 21 \times 1.3 \times 10^{-4} = 0.0027 \times 0.88 = 0.0024 \text{ in.}$$

$$\text{Fittings } 1.95 \times 0.88 = 1.72$$

$$4.19 \text{ velocity head} = 0.0073 \text{ in.}$$

$$\text{Available head} = 0.0200 \text{ in.}$$

$$\text{Pipe and radiator} = 0.0073$$

$$\text{Head for velocity} = 0.0127 \text{ in.}$$

$$\text{Velocity head} = \frac{0.0127}{4.19} \text{ in.} = 0.00304$$

$$\text{Head for 200 fpm} = 0.00249 \text{ in.}$$

$$\text{Velocity} = \sqrt{\frac{0.00304}{0.00249}} \times 200 = 221 \text{ fpm}$$

Since the pipe friction and register friction will be measured for a higher velocity, the velocity resulting will be taken as 215 fpm without further recalculation of pressure drop.

$$\text{Size of cool air duct} \frac{115}{1.152} \times \frac{144}{215} = 67 \text{ sq in. or } 8 \text{ in.} \times 9 \text{ in.}$$

$$\text{Size of hot air flue} \frac{115}{0.88} \times \frac{144}{215} = 87 \text{ sq in.} = 2\text{-}3\frac{1}{2} \text{ in.} \times 12 \text{ in.}$$

$$\text{Register face} \frac{87}{0.66} = 132 \text{ sq in.} = 10 \text{ in.} \times 12 \text{ in.}$$

The radiator surface for this room is found as

$$54 \times 2 \times \frac{8740}{9870} = 48 \text{ sq ft. Use } 50.$$

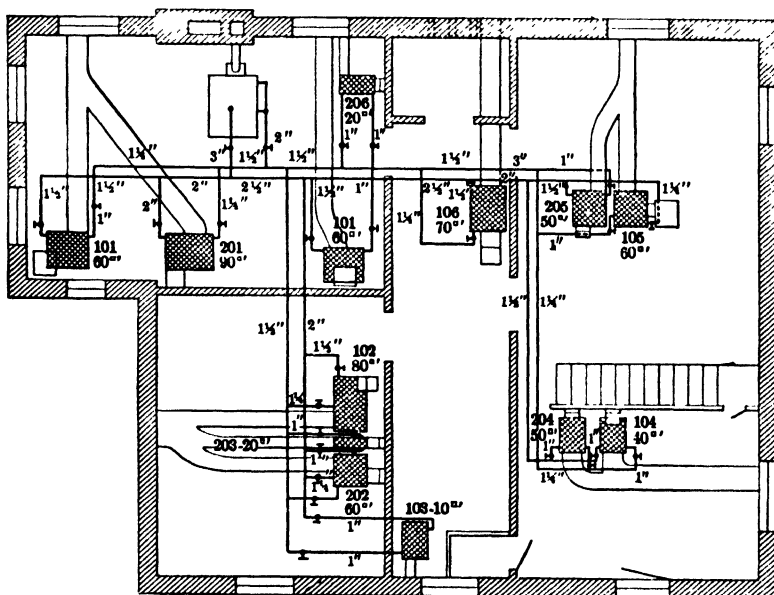


FIG. 156.—Cellar Plan with Indirect Radiators.

In the same way the other units are computed and tabulated. The steam pipes for supply and return are selected by finding the equivalent direct radiation (E.D.R.) and using this with the tables of Chapter VI. The value of E.D.R. is found by dividing the total heat by 240 Btu per hr per sq ft. The values from the table below are marked on the cellar plan, Fig. 156.

TABLE 8-7

INDIRECT GRAVITY HEATING RESIDENCE FOR L. Q. SMITH
AIR DELIVERY AT 140° F WITH 0° F OUTSIDE AIR

Room	Heat Loss	70° cfm	t_i	Total Heat	Cold Duct	Hot Flue	Register	Radiator, sq ft	E D R	Steam Branch
101	19 740	256	140	39 480	2-8×20		2-18×18	2-60	164	1½
102	14 275	186	140	28 550	1-8×33		1-20×24	1-80	119	1½
103	2 150	27	140	4 300	1-8× 5		1- 8× 8	1-10	18	1
104	6 480	86	140	12 960	1-8×15		1-15×15	1-40	54	1
105	10 100	133	140	20 200	1-8×23		1-18×18	1-60	85	1¼
106	12 250	163	140	24 500	1-8×28		1-20×20	1-70	102	1¼
201	15 140	203	140	30 588	1-8×17	3-3½×14	1-15×15	1-90	128	1¼
202	10 800	143	140	21 600	1 8×12	2-3½×14	1-12×15	1-60	90	1¼
203	3 600	48	140	7 200	1-8× 4	1-3½×10	1- 8× 8	1-20	30	1
204	8 400	111	140	16 800	1-8×10	2-3½×12	1-10×12	1-50	70	1¼
205	8 740	118	140	17 480	1-8× 9	2-3½×12	1-10×12	1-50	73	1¼
206	2 880	38	140	5 760	1-8× 4	1-3½× 8	1- 6× 8	1-20	24	1
Total	114 609			229 117				670	957	

All return branches may be ¾ in. unless 1 in. is the smallest size used by the designer

Steam lines, 0.05 # per 100 ft.

Steam main from boiler, 3 in.

Branches of main, 2 in., 1½ in., 2½ in., 2 in., 1½ in., 1 in.

PROBLEMS

1. A building requires 2500 Mbh to care for heat losses above 68° F. How much 70° F air per hr at an average inlet temperature of 95° F will be required? If all this air is taken from the outside at -5° F, how much additional heat will be required? Express this as a percentage of the heat for building loss. If one-half of the air is recirculated what additional percentage of the building loss will be needed?

2. A vertical flue 4 in. × 12 in. and 40 ft long is used to carry air. The air is discharged through a sharp right-angle bend and a grille. The air temperature in the flue is 110° F, and the building temperature is 70° F. Find the gravity circulating head in inches of water and in ounces of pressure per square inch. Find the diameter of equivalent round pipe. Find the quantity of air discharged through the pipe in cubic feet per minute if air is brought into the vertical pipe by pressure in the main and branch so that only the gravity head is used in vertical pipe friction and in loss at right-angle bend and grille.

3. How large a flue would be needed to supply the room of problem 2 with twice the amount of air of that problem? Of what thickness would the metal of the new duct be made?

4. The air discharged through a Venturi meter of 10 in. throat and 20 in. mouth is at 100° F and the inlet static pressure is of 1 in. of water above the atmosphere of 29.92 in. The difference in pressure is ½ in. of water. What volume of air in cubic feet per minute is entering the meter? What weight of air is discharged per second? How many watts of power would be supplied to a Thomas meter for this quantity of air if the temperature is raised from 100° F to 110° F? If a sharp-edged orifice of

10 in. diameter is used in the 20 in. pipe to measure this volume of air, what would be the drop in pressure at the orifice?

5. A grille of 80 sq in. gross area and 56 sq in. net area discharges air into a room with an average corrected anemometer reading of 1200 fpm. The air is at 85° F. How many cubic feet of 70° air are discharged per minute?

6. A room requires 10 Mbh above 72° F from an indirect pin radiator discharging air at 150° F. The flue from the return register grille is 30 ft long with 2 bends with R/W equal to 0.5. The friction loss in the radiator is to be taken from No. 61 flexitube of Table A-12. The vertical flue from the radiator is 15 ft long with a bend for which $R/W = 0.8$. There is a grille at the point of discharge. The temperature of the return air is 72° F. Find the sizes of inlet and outlet flues and the size of the radiator. Pins are 1 in. long.

CHAPTER IX

INDIRECT STEAM HEATING—PLENUM SYSTEM AIR CONDITIONING

To illustrate the computation for a plenum system of heating and ventilating, a school building shown in Figs. 157 and 158 will be examined, although in the practice of today such a building would probably be heated by the unit heaters. This is an 18 classroom building with a

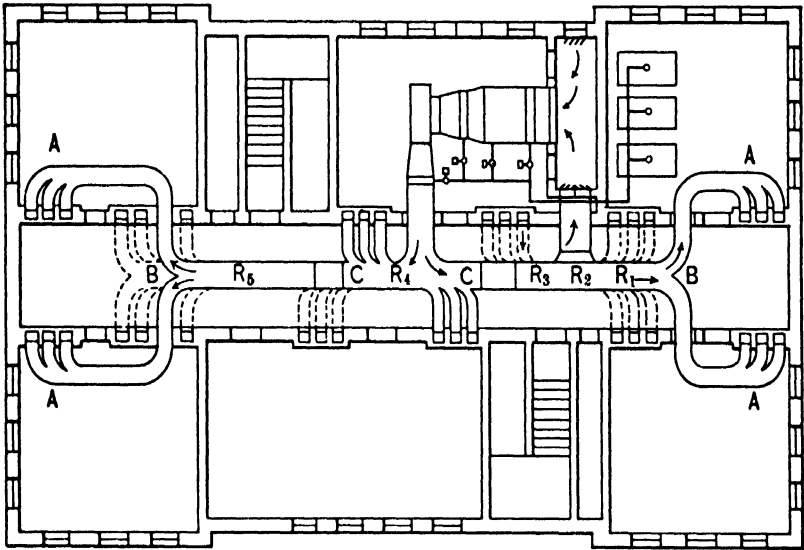


FIG. 157.—Double-duct System of Heating School.

coatroom adjacent to each. Offices are placed at the end of each corridor. Air is delivered from the inner wall of each room at an elevation of 9 ft through a long, narrow grille, and discharges through a register at the floor line through the coatroom. It then passes to the corridor and part is returned during occupancy to the basement for recirculation, the remainder passing to the roof by vertical ducts in the partition walls. Before occupancy in the morning all air is recirculated until the building is brought to the proper temperature. During the school sessions the recirculated air enters a chamber in which it mixes with the fresh air.

In this design, Fig. 160, the mixture is passed through a filter which

is intended to remove the finest dust and any insoluble matter which might pass through the washer.

The air is drawn by a fan through the filter at 325 fpm and, after tempering, through the washer and humidity-control chamber at 500 fpm,

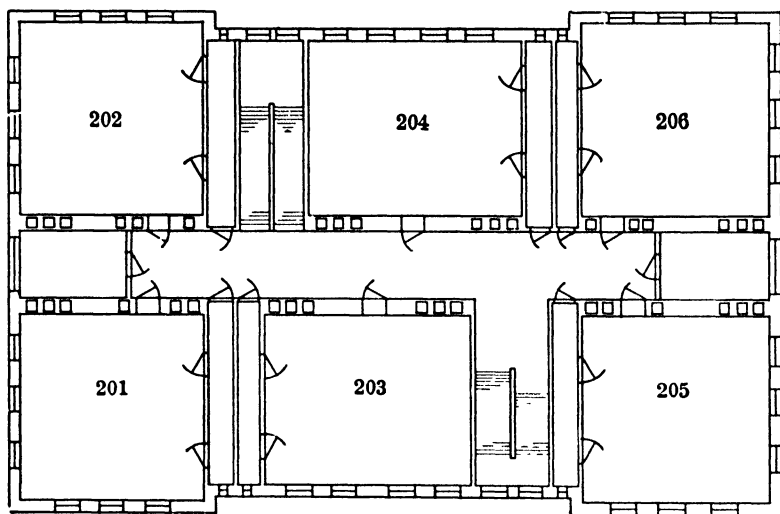


FIG. 158.—Second Story of School Building.

as well as a second tempering coil beyond the washer. This fan delivers part of the air through a heating coil and part through a by-pass to the dampers which permit the proper mixing of air at two temperatures to obtain any temperature between those of the two streams. The mixing is done near the heater in the single-duct system or at the base of the

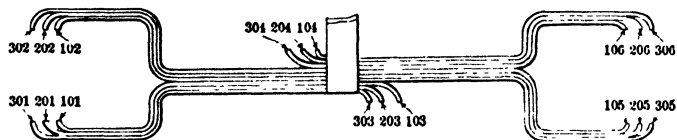


FIG. 159.—Single-duct System of Heating School.

flues in the double-duct. The single-duct system is shown in Fig. 159 and the double-duct in Fig. 157.

Fans may be placed in one of two locations. The blow-through arrangement is one in which the fan is placed at the air inlet, and the discharge, at relatively high velocity, must be reduced in speed in passing through heaters and washer and again increased in speed as it enters the duct systems. In the draw-through arrangement, the fan draws

the air at entering velocity through heaters and humidifier and delivers into the ducts at about the velocity from the fan, thus eliminating the loss due to changes in velocity. When part of the air is to be by-passed around a heater to get different temperatures for the room air, as in this problem, it is customary to use a blow-through installation at this point.

A study is made of the heat requirements of each room, according to the method of Chapter V, and the data sheets with equivalent air at 70° give the following table:

TABLE 9-1
REQUIREMENTS FOR SCHOOL NO. 3—RYDAL, N. Y.

First Floor				Second Floor				Third Floor			
Room	Occ.	Cfm	Mbh	Room	Occ.	Cfm	Mbh	Room	Occ.	Cfm	Mbh
101	49	1470	29 1	201	49	1470	29 0	301	49	1470	30 0
101A	.	.	1 8	201A	.	.	1 8	301A	1 8
102	49	1470	27 0	202	49	1470	26 0	302	49	1470	28 0
102A	1 5	202A	..	.	1 5	302A	..	.	1.5
103	49	1470	21 0	203	49	1470	20 0	303	49	1470	22 0
103A	.	.	1 5	203A	.	.	1 8	303A	.	.	1 8
104	49	1470	19.0	204	49	1470	18 0	304	49	1470	20 0
104A	.	.	1 5	204A	.	..	1 5	304A	.	.	1 5
105	49	1470	33 0	205	49	1470	32 0	305	49	1470	34 0
105A	.	.	1 8	205A	.	.	1 8	305A	.	.	1 8
106	49	1470	25 0	206	49	1470	24 0	306	49	1470	24 0
106A	1 8	206A	.	.	1 8	306A	1 8
Hall	.	.	10 2	Hall	.	.	5 0	Hall	.	.	4 0
Total	8820	174 2	Total	8820	164 2	Total	.	8820	172 2

This school has a total heated volume of 390,000 cu ft in an over-all cubage of 780,000 cu ft. The classrooms have 21 sq ft floor space and 250 cu ft per occupant.

In this design, 30 cfm of 70° F air at standard pressure, or 1800 cfm per occupant, is circulated through the rooms and 50% of this air is to be recirculated, giving 15 cfm of fresh air per occupant, which is above the amount required by the A.S.H.V.E. Code.

The rooms will be held at an effective temperature of 66° F, using the winter comfort line which will come from 70° dry bulb and wet bulb of 58° F, or 48% relative humidity (52 grains per pound, specific humidity). The system will be designed for outside conditions at 0° F with relative humidity of 50%.

The table gives the occupancy, the 70° air requirements and heat requirements of the walls, ceilings and floors, including air leakage, but no allowance has yet been made for sun effect or for occupancy. The total occupancy is 864 pupils and 18 teachers for the classrooms and a staff of 8 adult members. The total heat for the building demand is 510.6 Mbh. The total circulated air is 26,460 cfm.

The glass area on the south side is 1200 sq ft, and the net insulated wall area with a value of U of 0.115 is 4000 sq ft. The flat roof area is 12,000 sq ft, and the value of U with air space is 0.10.

The bodily heat given out during occupancy is

$$\begin{array}{rcl} \text{Pupils } 864 \times 250 \text{ Btu} & = & 216.0 \text{ Mbh} \\ \text{Adults } 26 \times 400 \text{ Btu} & = & 10.4 \\ \hline & & 226.4 \text{ Mbh} \end{array}$$

Of this, 20% is latent heat in the moisture from perspiration, leaving 181.1 Mbh for aid in caring for heat loss, although the full amount must be considered in the design of the heaters on account of the enthalpy of the mixtures.

Assume that thirty-six 40-watt lamps in the coatrooms and twenty 100-watt lamps in the halls are being used during the sessions of the school. These give a total of 3440 watts.

$$\text{Electric heat } 3440 \times 3.410 = 11.700 \text{ Mbh}$$

The sun effect on the south wall and roof is now computed:

$$\begin{array}{rcl} \text{Wall } 4000 \times 0.115 \times 23^\circ & \dots\dots\dots & 10.6 \text{ Mbh} \\ \text{Roof } 12,000 \times 0.10 \times 39^\circ & \dots\dots\dots & 46.8 \\ \text{Glass } 1200 \times 81 & \dots\dots\dots & 97.2 \\ \hline & & 154.6 \text{ Mbh} \end{array}$$

The collected data are as follows:

$$\begin{array}{rcl} \text{Outside air taken in at } 0^\circ (70^\circ) & \dots\dots\dots & 13,230 \text{ cfm} \\ \text{Recirculated air at } 70^\circ (70^\circ) & \dots\dots\dots & 13,230 \text{ cfm} \\ \text{Air delivered by fan } (70^\circ) & \dots\dots\dots & 26,460 \text{ cfm} \\ \text{Heat loss from building structure} & \dots\dots\dots & 510.6 \text{ Mbh} \\ \text{Electric energy heat} & \dots\dots\dots & 11.7 \text{ Mbh} \\ \text{Bodily heat (sensible } 181.1 \text{ Mbh)} & \dots\dots\dots & 226.4 \text{ Mbh} \\ \text{Sun effect} & \dots\dots\dots & 154.6 \text{ Mbh} \\ \text{Condition of air in rooms after supplying heat loss} & \left\{ \begin{array}{l} 70^\circ \text{ dry bulb} \\ 58^\circ \text{ wet bulb} \end{array} \right. & \end{array}$$

In planning the heating of this building it is to be realized that before occupancy in the morning 510.6 Mbh must be delivered into the rooms

to care for the heat losses from the structure, the air to each room being delivered at the temperatures given in the table below so as to supply the heat to that room by the amount of air being discharged. If it is desired to heat the building to a proper temperature because of cooling at night, then higher temperatures could be given to the incoming air during this period before the school session begins. During this period, 100% of the air is recirculated and the water spray is omitted from the washer. After the session begins, the sensible heat from the occupants and the lights, as well as the sun effect on a clear 0° F day, will reduce the demand for heat beyond that for the fresh air from 510.6 to 163.2 Mbh. If the sun does not shine, the heat becomes 317.8 Mbh, and if no electric lights are in use, 329.5 Mbh will be required. The temperature of the air entering the room, so that on cooling to 70° it will supply the heat loss from that room, is computed for an unoccupied room and for the room when occupied. At this latter time the sensible bodily heat from the occupants reduces that needed from the air for the heat loss.

TABLE 9-2
TEMPERATURES, t_i , OF INLET AIR, DEGREES F

First Floor			Second Floor			Third Floor		
Room	Unocc.	Occ.	Room	Unocc.	Occ.	Room	Unocc.	Occ.
101	88.4	81.5	201	88.3	81.5	301	88.9	81.1
102	87.0	79.2	202	86.4	78.6	302	87.6	79.8
103	83.2	75.4	203	82.6	74.8	303	83.9	76.0
104	82.0	74.0	204	81.3	73.5	304	82.6	84.8
105	90.8	83.0	205	90.1	82.4	305	91.6	83.6
106	85.7	78.0	206	85.1	77.3	306	85.1	77.3

For operation during occupancy the temperatures of the entering air will vary, 83.6° F for room 305 to 73.5° F for room 204. The higher temperatures, 91.6° F and 81.3° F, for these rooms before occupancy, occur with 100% of recirculated air, and the heaters which care for occupancy with some zero degree air will be sufficiently large for these temperatures.

On account of the variation of humidity the air conditions at the various points must be computed.

To determine the volume of 1 lb of air and its associated moisture of relative humidity, ρ , the following equations * are used:

* See page 33 for derivation.

$$v_p = \frac{v_s - v_d}{v_s - \rho(v_s - v_d)} \quad \text{or approximately} \quad (9-1)$$

$$v_p = v_d + \rho(v_s - v_d) \quad (9-2)$$

v_s = volume of 1 lb of air associated with moisture to saturate, cubic feet per pound of air

v_d = volume of 1 lb dry air, cubic feet per pound of air

v_p = volume of 1 lb of air associated with moisture to give relative humidity ρ , cubic feet per pound of air

Volume of 1 lb of air 0° F – 50% relative humidity

$$= 11.58 + \frac{1}{2} \times 0.01 = 11.585 \text{ cu ft}$$

Volume of 1 lb of standard 70° dry air = 13.35 cu ft

Volume of 1 lb of air at 70° F dry bulb and 58° wet with 29.92 in. is

$$V = 13.35 + 0.48 \times 0.34 = 13.51 \text{ (exact 13.51)}$$

Dew point of air at 70° dry bulb and 58° wet bulb with a pressure of 29.92 in. is 49.0° F

Weight of standard 70° dry air in pounds per minute is

$$W = 13,230 \times \frac{1}{13.35} = 990 \text{ lb per min}$$

$$\text{Moisture per pound of fresh air} = \frac{5.47}{2} = 2.7 \text{ grains}$$

$$\text{Moisture in fresh air, grains per minute} = 990 \times 2.7 = 2673 \text{ grains}$$

Weight of dry air delivered to room, pounds per minute

$$2 \times 990 = 1980$$

Moisture per pound of air delivered from chart = 52.0 grains

Moisture in air delivered, grains per minute

$$= 1980 \times 52.0 = 102,960 \text{ grains}$$

Perspiration given off per person per minute (assume 50% on account of pupils)

$$= 0.50 \times \frac{650}{60} = 5.4 \text{ grains}$$

Moisture returned with recirculated air, grains per minute

$$= \frac{1}{2}(102,960 + 890 \times 5.4) = 53,880 \text{ grains}$$

Moisture in recirculated air per pound of air

$$\frac{53,880}{990} = 54.4 \text{ grains}$$

Moisture added in humidifier = $102,960 - (2673 + 53,880)$ grains

$$= 46,407 \text{ grains per min} = 6.63 \text{ lb per min}$$

$$= 398 \text{ lb per hr}$$

The heating coils will be applied for the following duties:

First tempering coil:

Heat the mixture of 990 lb of air per min at 0°F and 50% relative humidity and 990 lb of air per min at 70°F with 54.4 grains of moisture to a temperature such that on cooling to 49.0°F it will abstract moisture from water at 49.0°F to saturate it at this temperature.

Second tempering coil:

To heat 1980 lb of air per min from saturation at 49°F to a temperature below the lowest temperature of inlet air needed in any room.

Third heating coil:

To heat part of the air to some temperature above that of the highest temperature of the inlet air.

These will be considered in detail:

First Tempering Coil. The condition after mixing the fresh air and recirculated air must be found.

Air at zero degrees with 50% relative humidity contains 2.7 grains of moisture per lb (0.5×5.47) and has an enthalpy value of 0.4 Btu per lb.

Air at 70°F , 29.92 in. pressure and 54.4 grains per lb will have a wet-bulb temperature of 58.6°F from the Carrier chart.

From Table A-2, $h = 25.55$.

Since the weights of the recirculated and fresh air are equal, the enthalpy per pound of the mixture and the moisture per pound of air will be the arithmetic means of the respective values

$$\frac{2.7 + 54.4}{2} = 28.6 \text{ grains of moisture per lb air}$$

$$\frac{0.4 + 25.55}{2} = 12.97 \text{ Btu per lb}$$

The wet-bulb temperature for this enthalpy is 34.9° , and from the Carrier chart the dry-bulb temperature for 28.6 grains on this wet-bulb line is 35.5°F .

The enthalpy of the saturated air at the dew point, 49.0°F , is 19.78 Btu per lb, and the condition of air containing this enthalpy with 28.6 grains of moisture would be a wet-bulb temperature of 49.0°F and a dry-bulb temperature of 62.8°F . If now the mixed air at 35.5°F dry-bulb temperature and 28.6 grains per pound (enthalpy, 12.97 Btu per lb) is heated to 62.8°F dry-bulb temperature (enthalpy, 19.78 Btu per lb) and then passed through water at 49.0°F , it will be brought to a saturated condition at this temperature. The evaporation of 6.63 lb of water per min by the lowering of the temperature from 62.8° to 49.0°F will increase the moisture content from 28.6 grains per lb to 52 grains per lb.

Air from 35.5° to 62.8° will require a face velocity of 660 ft per min with No. 61 Flexitube Aerofin heater, as shown by Table A-11 by interpolation. For these conditions the condensation is 2.19 lb per hr per sq ft.

$$\text{Heat from coils} = 1980 \times 60(19.78 - 12.97) = 809 \text{ Mbh}$$

$$\text{Face area required} = \frac{1980 \times 13.35}{660} = 40.0 \text{ sq ft}$$

$$\text{Heating surface required} = \frac{809,000}{2.19 \times 960.6} = 384 \text{ sq ft}$$

(960.6 = latent heat of evaporation at 227° F)

From Table A-10 of physical data this requires four 5 ft No. 61 units giving 41.44 sq ft of net face area and 384 sq ft of heating surface. The space requirements would be 116 in. \times 68½ in. \times 10 in.

Second Tempering Coil. The air from the washer at 49.0° F will now be heated to 73° before entering the fan. This will require (by extrapolation) one set of tubes of No. 61 Flexitube Aerofin heaters at 800 fpm face velocity with a condensation of 2.26 lb per sq ft of heating surface per hour.

The enthalpy of 73° air with 52 grains of moisture of 59.2° F wet bulb is 25.95 Btu per lb.

$$\text{Heat from coils} = 1980 \times 60(25.95 - 19.78) = 733 \text{ Mbh}$$

$$\text{Face area required} = \frac{1980 \times 13.35}{800} = 33.1 \text{ sq ft}$$

$$\text{Heating surface required} = \frac{733,000}{2.26 \times 960.6} = 337 \text{ sq ft}$$

This requires three 6 ft No. 61 Flexitube Aerofin units giving 37.35 sq ft of net face and 345 sq ft of heating surface.

These would require 87 in. \times 80½ in. \times 10 in.

Third Heater. The heater would use some of the air at 73° and increase its temperature to a point above 83.6° F. A temperature of 93.0° F will be taken which by extrapolation requires a face velocity of 800 fpm with No. 61 unit and gives a condensation of 1.98 lb per sq ft per hr.

The amount of air to be handled at 93° and at 73° will be such that when these cool to 70° F the heat given up will be equal to the heat loss from the building, less the sensible heat from the occupants. The heat loss from the building is 510.6 Mbh. The sensible heat production from the occupants is 181.1 Mbh. The air at 70° and 52 grains of moisture has an enthalpy value of 25.16 Btu, the air recirculated contains 25.55 Btu.

The air at 93° F with 52 grains of moisture has a wet-bulb temperature of 66.2° F and an enthalpy of 31.01 Btu per lb. The equation for heat loss becomes:

$$M(31.01 - 25.16) + (1980 - M)(25.95 - 25.16) = \frac{510.6 - 181.1}{60} \times 1000$$

$$M = 776 \text{ lb per min}$$

$$\text{Heat from coils} = 776 \times 60(31.01 - 25.95) = 235.6 \text{ Mbh}$$

$$\text{Face area required} = \frac{776 \times 13.35}{800} = 12.9 \text{ sq ft}$$

$$\text{Heating surface required} = \frac{235,600}{1.98 \times 960.6} = 124 \text{ sq ft}$$

This requires two $3\frac{1}{2}$ ft No. 61 Flexitube Aerofin units giving a net face area of 14.44 sq ft and 136 sq ft of heating surface.

These require 58 in. \times 50 $\frac{1}{2}$ in. \times 10 in.

It would have been possible to add the heat necessary to care for insulation loss of the duct if thought desirable. This loss might cause a drop of $\frac{1}{2}^{\circ}$ F which would be cared for by an increase of temperature of this amount and an increase in enthalpy of 0.12 Btu per lb.

These heats will be assembled for the purpose of checking results.

HEAT DEMAND

From building loss	510.6 Mbh
From exhausted air, $990 \times 60(25.55)$	1517.7
	<hr/>
	2028.3 Mbh

HEAT SUPPLY

From fresh air, $990 \times 60 \times 0.4$	23.8 Mbh
From occupants (total)	226.4
From 1st heater	809.0
From 2nd heater	733.0
From 3rd heater	235.6
	<hr/>
	2027.8 Mbh

Steam Pipe Sizes. The sizes of the pipes supplying the various units are taken from the tapping list of the manufacturer, as given on page 129. The other pipes would be computed as shown in Chapter VI.

General Arrangement. The heater, washer and fan would be arranged as shown in Fig. 160.

The fresh air inlet for 11,450 cfm at 0° F and for 700 fpm is of 48×48 in. size, Fig. 157, and is controlled by a series of damper vanes which are closed when all the air is to be recirculated. The return duct inlet from the building is 60×72 in. for 26,460 cfm of 70° air with a velocity of 880 fpm. It is equipped with dampers operating over one-half of the area to shut off this part when only 50% of the air is to be recirculated. The other half of the return air inlet is equipped with damper vanes which may be adjusted to control the mixture of recirculated air.

The frontal area of the filter is computed for a velocity of 325 fpm and the spray chamber cross-section is such that the air moves at 500 fpm. The length of the washer is about 10 ft.

As the outside temperature rises, less heat will be required from all the heaters as the outdoor heat will contain more moisture and enthalpy. Moreover, the heat loss from the building will be less. Since the quantity of circulated air remains constant, thermostats must regulate the supply of heat to the coils. In general, a thermostat controls the admission of steam to the first heater so that its air discharge at the end of the washer is at 49.0°F . A thermostat in the basin at the bottom of the spray chamber keeps the water at 49.0°F . A third combined out-

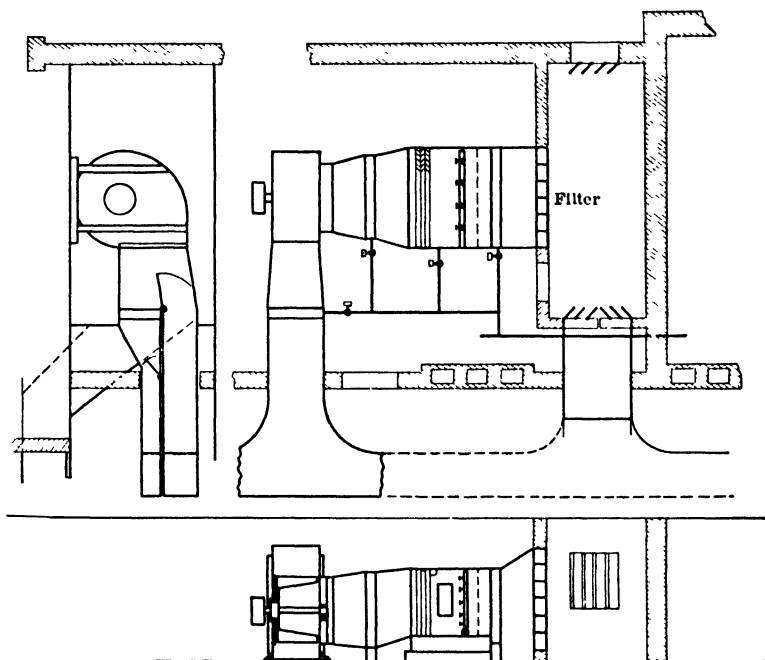


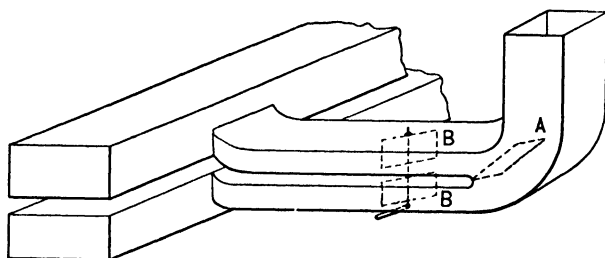
FIG. 160.—General Arrangement of Air Conditioner.

side thermostat and fan inlet thermostat controls the second heater steam to keep the air inlet to the fan at 73°F in zero weather and lower at higher outside temperatures. A fourth would control the steam to the third coil so that its outlet would never be above 93°F , and would reduce it for higher outside temperatures.

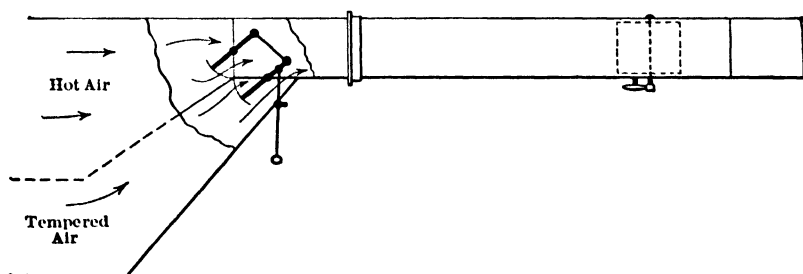
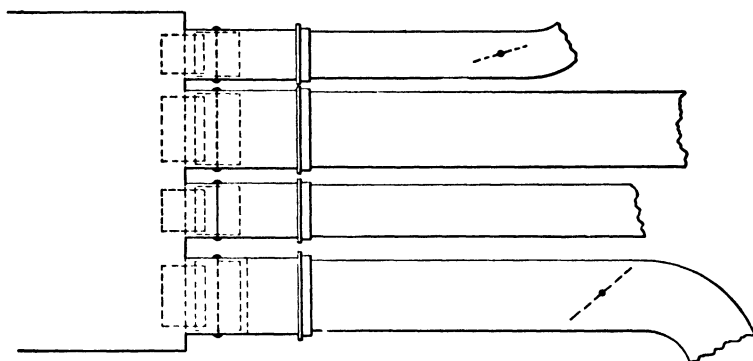
The mixing dampers for each room, Fig. 161, vary the temperature of the room air to care for the heat loss. They are operated by hand or by room thermostats.

Hot water may be used to humidify the air if the discharge from the first tempering coil is fixed at 49.0°F , and water is introduced into the washer above this temperature so that the water in giving up its mois-

ture is cooled and a thermostat in the outlet to the washer controls a circulating spray water heater to maintain the outlet temperature at 49.0°F .



Double-duct System



Single-duct System

FIG. 161.—Mixing Dampers.

Duct System. The double-duct system is designed for this building, Fig. 157, in which 1204 lb of air per min at 73°F are carried through one distributing duct and 776 lb of air per min at 93°F are carried through another. The flue for any particular room receives amounts

of air from each duct to produce the desired inlet temperature, and the dampers regulating this mixture are under hand or thermostatic control. After fixing the general arrangement of the ducts and flues the velocities in the various parts of the system are assumed and, from Fig. 148, the sizes of circular ducts or flues are selected to carry the actual cfm of realtive density x .

$$\text{Actual cfm} = \frac{\text{Cfm at } 70^{\circ} \text{ F}}{x} \quad (9-3)$$

If the chart, Fig. 148, is not used the diameter may be found by

$$\frac{\pi d^2}{4} = \frac{\text{Cfm at } 70^{\circ}}{xV} \quad (9-4)$$

The longest circuit to room 306 is first considered. The velocity head at any point in inches of water is computed as

$$h_i = x \left(\frac{V}{4005} \right)^2 \quad (9-5)$$

Since this head is equal to the loss from the friction in an elbow equivalent approximately to that from 50 diameters of length, the equivalent length of pipe with friction from pipe, elbows and fittings may be found.

The total equivalent length of pipe, fittings, grilles and final velocity head is found in feet, and the losses in filters, washers and heaters are known in inches. After the designer assumes the total loss in head desired through the longest circuit, the loss per foot or per 100 ft in inches of water can be found. This quantity is then fixed and the sizes of pipes are selected from Fig. 148 for the given air quantities with this loss per 100 ft. Equivalent rectangular ducts for this loss and quantity are found by Table A-18, or by the formula on page 236, and the friction loss in various parts of the circuit can be determined in inches of water. The actual velocities for these rectangular cross-sections are computed to determine the losses in fittings and elbows, as these are given in terms of velocity heads. The sum of the losses will give the maximum total actual head which must be cared for by the fan.

Instead of this method of equal drops per 100 ft, the Buffalo Forge Company suggests that, in long pipe systems for industrial buildings, the velocity assumed at the fan be decreased in the main or trunk line after each branch so that the increase in pressure head due to the decrease in velocity head will care for the friction losses in the next section of the main. Thus, the velocity head in section 1 will equal the

velocity head and losses in section 2 of the trunk which extends from section 1 to section 3.

$$\frac{V_1^2}{2g} = \frac{V_2^2}{2g} + 4f \frac{L}{D} \frac{V_2^2}{2g} + nc \frac{V_2^2}{2g} \quad (9-6)$$

$$V_2 = \frac{V_1}{\sqrt{1 + 4f \frac{L}{D} + nc}} \quad (9-7)$$

n = number of fittings, including diffusers;

c = loss at fitting in terms of velocity head.

This is then repeated for sections 3, 4, 5, and so on. In all cases then there must be sufficient velocity head to care for the last section, so that it might be better to start at the last section and work toward the original one.

As a first approximation in the design for this school, the temperature will be taken as 77.3° F (t , for room 306) for the complete circuit to room 306. The velocities assumed will be 250 fpm for outlet to room, 750 fpm for flue and 1000 fpm for ducts. Reference is made to Figs. 157 and 148.

Actual amount of air for room 1470 ÷ 0.986 = 1490 cfm

Size of equivalent pipe for outlet $\sqrt{\frac{1490 \times 144}{250 \times 0.785}} = 33$ in.

Size of equivalent pipe for flue at 750 fpm (Fig. 148) = 18 in.

Equivalent length for loss at outlet $1.75 \times \frac{33}{12} \times 50 = 240$ ft

Equivalent length for loss at elbow and enlargement $1.75 \times \frac{18}{12} \times 50 = 130$ ft

Length from flue to basement = 35 ft

Equivalent length for loss in two elbows ($R/D = 1.0$)

$$2 \times 0.26 \times \frac{18}{12} \times 50 = 39 \text{ ft}$$

Size pipe from *A* to *B* for 3×1490 or 4470 cfm at 1000 fpm = 30 in.

Equivalent length for loss in two elbows ($R/D = 1.5$)

$$2 \times 0.17 \times \frac{30}{12} \times 50 = 42.5 \text{ ft}$$

Length *A* to *B* (nothing allowed for contraction) = 25.0 ft.

Size pipe from *B* to *C* for 2×4470 cfm at 1000 fpm = 40 in.

Equivalent length for converging section $0.05 \times \frac{40}{12} \times 50 = 8$ ft

Length *B* to *C*..... = 30 ft

Size pipe from *C* to junction for 3×4470 cfm at 1000 fpm = 54 in.

Equivalent length for loss in elbow ($R/D = 1.0$).... $0.26 \times \frac{54}{12} \times 50 = 59$ ft

Length *C* to junction..... = 5 ft

Size pipe from junction to heater for 6×4470 cfm..... = 72 in.

Length to heater..... = 10 ft

Total equivalent length..... = 623.5 ft

Loss in heater at 800 fpm or by-pass = 0.170 in.

Loss in second tempering coil at 800 fpm..... = 0.170

Loss in washer at 500 fpm..... = 0.250

Loss in first tempering coil at 660 fpm..... = 0.108

Loss in filter..... = 0.150

Loss at entrance and velocity head at 700 fpm 1.5×0.03 = 0.045

Total..... = 0.893 in.

The friction of 623 ft and 0.893 in. would be the total dynamic head as losses have been considered in converging devices. If the total static head is taken as 1 in. of water, with 0.06 in. as the velocity head, the head loss in the pipes will be

$$1 \text{ in.} + 0.06 - 0.893 = 0.167 \text{ in.}$$

This will give

$$\frac{0.167}{6.235} = 0.03 \text{ in. per 100 ft}$$

On account of the approximations used so far in the design the loss used for actual design will be 0.05 in. per 100 ft, and if necessary a change will be made after an accurate computation.

Final Determination of Pressure in Inches.

Grille loss and velocity head at 250 cfm for air at 77.1°F

$$1.75 \times 0.0039 \times 0.986 \text{ (Tables 8-2 and 8-4)} \dots = 0.0067 \text{ in.}$$

Rectangular flue for same friction as that of 18 in. pipe

$$\text{from Table A-18 or Fig. 149} \dots = 17 \text{ in.} \times 17 \text{ in.}$$

$$\text{Velocity in flue } \frac{1490 \times 144}{17 \times 17} \dots = 742 \text{ fpm}$$

Velocity head..... = 0.0345 in.

Loss in enlargement, 3 elbows and damper at bottom

$$(0.75 + 3 \times 0.26 + 0.5) 0.0345 \times 0.986 \dots = 0.0695 \text{ in.}$$

$$\text{Loss in 35 ft of pipe } \frac{35}{100} \times 0.05 \times 0.986 \dots = 0.0173 \text{ in.}$$

A to B. In this section, air at 93°F is carried in one duct and at 73°F in the other. The inlet air temperature to room 306 will require 300 cfm from the 93° trunk, and 1160 cfm from the 73°F trunk. In the same way rooms 106 and 206

will be examined, the total amount from the three flues at point *A* being 1000 cfm from the hot trunk and 3410 cfm from the tempered air trunk. Both of these are equivalent volumes at 70° F. These, for a loss of 0.05 in. per 100 ft, require 16 in. and 28 in. pipes respectively, for which the rectangular pipes with equal friction loss are 8 in. × 36 in. and 19 in. × 36 in. The velocities in these at actual temperatures are 503 fpm and 751 fpm for which the velocity heads are 0.0157 in. and 0.0350 in.

$$\begin{aligned}\text{Loss in 2 elbows} & \dots 2 \times 0.18 \times 0.0157 \times 0.958 = 0.0054 \\ & 2 \times 0.18 \times 0.0350 \times 0.994 = 0.0125\end{aligned}$$

$$\text{Loss in 25 ft pipe} \dots \dots \frac{25}{100} \times 0.05 \times 0.958 = 0.0120$$

$$\frac{25}{100} \times 0.05 \times 0.994 = 0.0125$$

In the same way from *B* to *C* there will be 3200 cfm in the hot trunk and 5610 in the tempered trunk requiring 24 in. and 32 in. round pipes for 0.05 in. per 100 ft for which the rectangular pipes for the same friction loss and quantity will be 12 in. × 48 in. and 20 in. × 48 in. The velocity heads in these will be 0.0418 in. and 0.0547 in.

From *C* to the junction the sizes will be 15 in. × 48 in. and 27 in. × 48 in. with velocity heads of 0.0421 in. and 0.0550 in. The ducts to the heater will be 21 in. × 60 in. and 36 in. × 60 in. with velocity heads of 0.0582 in. and 0.0965 in.

The total drop will now be computed.

HEAD REQUIREMENTS IN INCHES

	Hot	Tempered
Velocity head at discharge and loss in grille ...	0.0067	0.0067
Enlargement, 3 elbows and damper	0.0695	0.0695
35 ft pipe	0.0173	0.0173
25 ft pipe (<i>A</i> to <i>B</i>)	0.0120	0.0125
2 elbows	0.0054	0.0125
30 ft pipe (<i>B</i> to <i>C</i>)		
($\frac{3}{2} \times 0.0120$ and $\frac{3}{2} \times 0.0125$)	0.0144	0.0150
Converging section		
(0.05×0.0418 and 0.05×0.0547)	0.0021	0.0027
5 ft pipe (<i>C</i> to junction)		
($\frac{5}{2} \times 0.0120$ and $\frac{5}{2} \times 0.0125$)	0.0024	0.0025
1 elbow <i>R/D</i> = 1 rectangular		
(0.018×0.0421 and 0.018×0.0550)	0.0007	0.0010
12 ft pipe (junction to heater)		
($\frac{1}{2} \times 0.0120$ and $\frac{1}{2} \times 0.0125$)	0.0048	0.0050
Heater or by-pass	0.1700	0.1700
Second tempering coil	0.1700	0.1700
Washer	0.2500	0.2500
First tempering coil	0.1080	0.1080
Filter	0.1500	0.1500
Entrance or return ducts	0.0450	0.0450
	<hr/>	<hr/>
	1.0283	1.0377

This is sufficiently close to 1.09 in. assumed as the dynamic pressure so that the sizes used are correct.

The flues to each floor of section *A* have the same drop in pressure but the lengths are 35 ft, 21 ft and 7 ft respectively. For this reason those to the second and first floors could be made smaller, as shown in Chapter VIII.

$$D_2/D_1 = \sqrt[5]{L_2/L_1} \quad (9-8)$$

$$D_2 = 17 \sqrt[5]{\frac{21}{35}} = 15.5 \text{ in.}$$

$$D_1 = 17 \sqrt[5]{\frac{7}{35}} = 12.3 \text{ in.}$$

The flues for section *C* have more head causing flow because of the friction drop from *C* to *A*. This amounts to 0.0339 in. (0.0120 + 0.0054 + 0.0144 + 0.0021), giving a total loss for the flue of 0.0512 in. in place of 0.0173. Hence,

$$D_2 = D_1 \sqrt[5]{p_1/p_2} \quad (9-9)$$

$$D_{c_2} = 17 \sqrt[5]{0.0173/0.0512} = 13.6 \text{ in.}$$

$$D_{c_2} = 15.5 \sqrt[5]{0.0173/0.0512} = 12.4 \text{ in.}$$

$$D_{c_1} = 12.3 \sqrt[5]{0.0173/0.0512} = 9.9 \text{ in.}$$

Although these sizes could be used for the flues to the different floors at section *C* on account of the available pressure, they are usually made of the same size as those of *A* because of the desire to maintain the flue velocity of 750 fpm. Extra resistance is placed in the line by a regulating damper. Of course for very long flues or ducts it might be advisable to use the smaller size for economy in materials.

Following the design of the delivery duct and flues the same sizes will be used for the return flues to the basement and attic. The areas of the return ducts will be of the combined areas of hot and tempered air ducts as they return all the air to the inlet for recirculation of 100% of the air before school sessions.

After occupancy only 50% of the air is recirculated, and this reduces the return losses to $\frac{1}{4}$ of the 100% recirculation value.

The sizes of these ducts are as follows:

TABLE 9-3
SIZES OF DUCTS

	Heated Air	Tempered Coil
Heater to junction.....	21 × 60	36 × 60
Junction to <i>C</i>	15 × 48	27 × 48
<i>C</i> to <i>B</i>	12 × 48	20 × 48
<i>B</i> to <i>A</i>	8 × 36	19 × 36
		Mixed Air
Supply flues		17 × 17
Return flues.....		17 × 17
Return trunk <i>R</i> ₁ and <i>R</i> ₂		25 × 60
Return trunk <i>R</i> ₃		37 × 60
Return trunk <i>R</i> ₄		50 × 60

These supply ducts are made of metal thicknesses given in Table A-19 with proper stiffeners and standing joints. At intervals cleanout doors are formed in ducts. The dampers for balancing flow to various rooms should be arranged with set screws to hold them in final settings. If covering is placed on ducts and flues, this should be of proper thickness to hold the heat loss at the point assumed in the determination of heat. The return trunk will be formed of masonry below the floor, being connected to the mixing chamber by an inclined metal duct.

In place of this double-duct system with mixing at the base of the flues, which has been chosen to illustrate the design of this system, a simpler design and one which is more often used in practice, the single-duct system, Fig. 159, could have been used. In this the mixing takes place just beyond the heater, Fig. 160, and each flue and its duct are designed to carry air of a definite temperature from the mixing point to the room. The longest duct and flue determine the pressure needed at the mixing point for the whole system. The design of each single circuit is simple because it carries air of the same weight and temperature throughout its length, although the general method of procedure is that used for the double-duct system.

The fan must now be designed to produce the necessary suction and delivery pressures.

For suction:

Entrance of cold air	0 0450 in.
(This is less than return system $\frac{1}{4} \times 0.1437 = 0.0359$)	
Filter	0.1500
Tempering coil	0 1080
Washers	0 2500
Tempering coil	0 1700
	<hr/>
	0 7230 in.

For delivery:

Heating coil or by-pass	0.1700 in.
Piping system	0.1447
	<hr/>
	0.3147 in.

The total head produced by the fan will be 1.0377 in., which is the pressure including the velocity heads at each end. The dynamic head on the fan will be used as 1.0297 in. excluding the final velocity head.

Fans. The centrifugal fan consists of a rotor mounted on a shaft which turns within a casing. The blades of the rotor are attached to rings of sheet metal carried by arms, Fig. 162, or to a disc at one end and a ring at the other, Fig. 163. The blades may be flat radial sheets or curved members, as shown in the figures. At times the discharge

edges of the curved blades are turned in the direction of the peripheral motion, forming forward-curved blades, or they may be turned at this point in a direction opposite to that of peripheral motion, backward-curved blades, Fig. 164. In all cases the entrance side of the blade

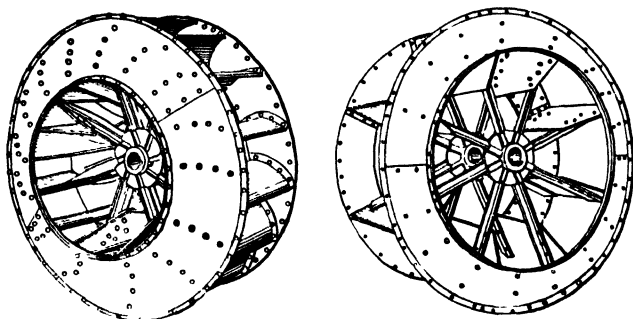
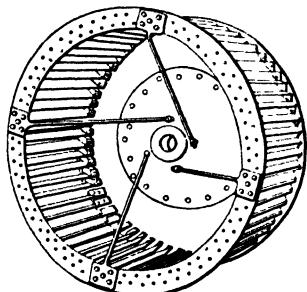


FIG. 162.—Curved Blade and Radial Blade Fans.

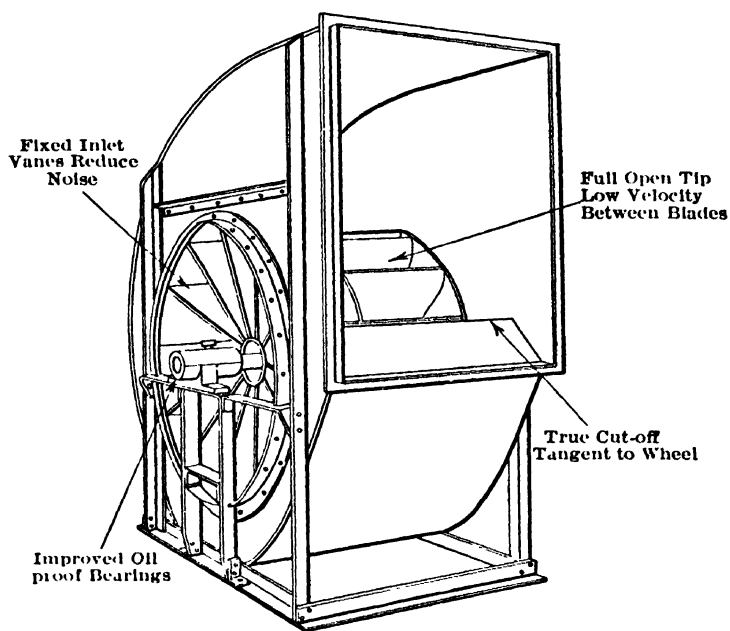
should be bent forward so as to receive the air with little interference. Each of these shapes has certain peculiarities which are of value under definite conditions. Thus, the forward-curved blade requires a much lower peripheral speed than the backward-curved blade for a given velocity of discharge, and hence it would be used with low-speed motors. Again, these two forms of blade give different pressure and power characteristics which make the backward blade preferable for installation of fans in multiple.



Courtesy of American Blower Co.

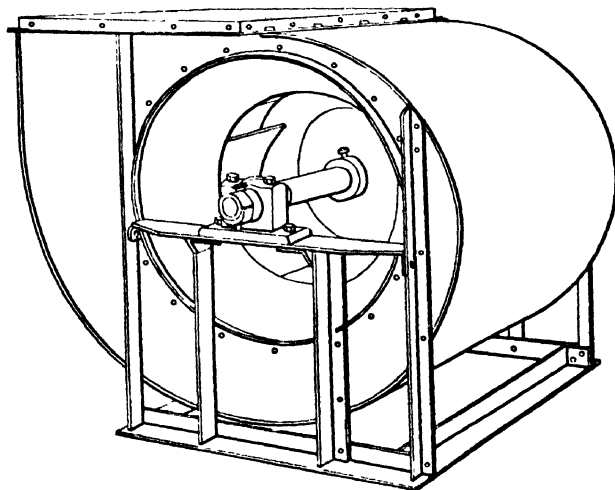
FIG. 163.—Sirocco Fan.

Of the fans in common use today, three are shown: the Sirocco fan of the American Blower Company, Fig. 163; the Limit Load Conoidal fan, Fig. 164, of the Buffalo Forge Company; and the Silent Vane fan, Fig. 165, of the B. F. Sturtevant Company. In the latter figures the rotor is seen within the scroll casing. This is so shaped around the rotor that a certain part of the velocity head is efficiently changed into pressure head. The Silent Vane fan illustrated has inlets on each side, being a double-width, double-inlet fan, while the Limit Load Conoidal is a single-width, single-inlet fan. The inlets of the former fan are open with a stream-line converging mouth-piece; the latter has a set of fixed inlet blades directing the air into the center of the rotor in such a way that losses and noises are reduced.



Courtesy of Buffalo Fan Co.

FIG. 164.—Limit Load Conoidal Fan.



Courtesy of B. F. Sturtevant Co.

FIG. 165.—Silent Vane Fan.

Fans are built with outlets discharging horizontally, vertically and in an inclined direction. There are sixteen standard arrangements of which eight are shown in Fig. 166. The nominal directions of rotation associated with these are obtained when the fan is viewed from the driven side. The remaining eight arrangements are similar to those shown, except that the outlets are on the side opposite that shown, as for every type there is clockwise and counterclockwise rotation.

The blades of the Limit Load Conoidal fans are built with a variation in radial depth, tapering from a narrow blade at entrance to a deeper blade at point of attachment to the disc. The amount of inlet air traveling parallel to the shaft becomes less and less as the disc is approached so that less clear space at center is needed, and this increase in radial

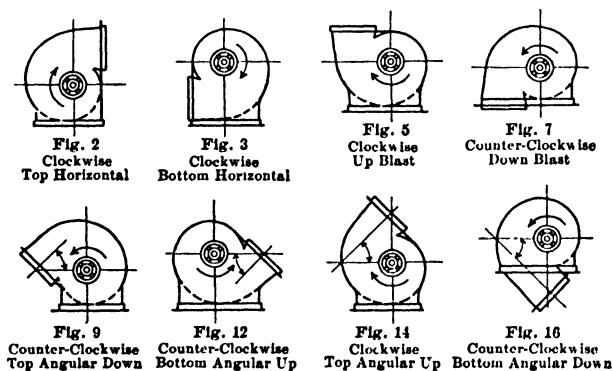


FIG. 166.—Standard Fan Arrangements.

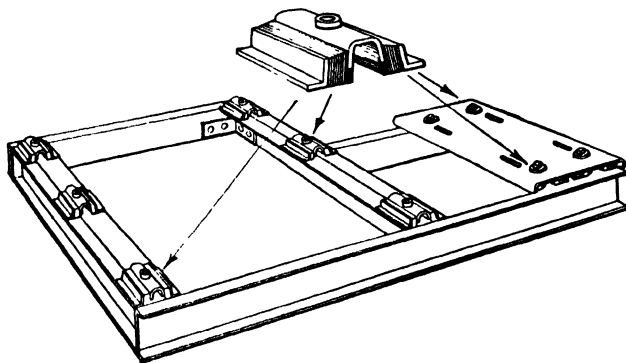
length of blade results in greater fan efficiency for the longer parts of the blade.

Several standard arrangements of bearings are used with fans. In some, bearings are on each side of the casing; and in others, the shaft with the fan overhangs the bearing, giving an inlet free from such obstruction. The drive pulley may be overhung or the pulley may be placed between two bearings. The fan frame is built at times with an extension base for the motor or turbine mounting, the shaft being furnished with a flange coupling.

To reduce noise and vibration, the Buffalo Forge Company has developed its so-called silent floating base, Fig. 167, which extends under fan and motor frames, supporting them by small rubber strips which are attached to the metal of the supporting pad and extensions on the base, so that this rubber is in shear and there is no direct metal connection between fan and motor frames and base. Fans and motors may

be mounted on materials such as cork, felt, rubber or even springs to prevent vibration and noise passing into the building structure. These flexible supports should be loaded up to their allowable capacity rather than being of large area with small unit load.

The outlet of the fan casing is connected to the duct by means of canvas to reduce noise and vibration which might be carried by the



Courtesy of Buffalo Forge Co.

FIG. 167.—Floating Fan Base.

duct system, and ducts may be lined with a sound-absorbing material, or with an inner metal lining having perforations to reduce noise. When noise from a fan room is conducted to other parts of a building, sound-absorbing material must be placed on the walls and ceiling of the room.

Propeller fans, Fig. 168, are used for ventilating kitchens, stores, gymnasiums, laboratories, restaurants and similar places by removing air from the space and discharging it outside. They handle large quantities of air with almost no static pressure.

Fan Characteristics. At a given speed of rotation, the static pressure produced by a fan varies with the amount of air flowing. The relation between pressure or required horsepower and volume, as determined by test, is shown in Fig. 169 for both the backward-curved Limit Load Conoidal fan and a forward-curved

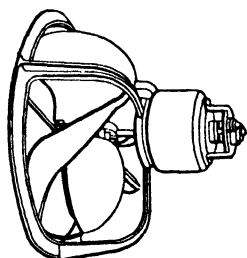


FIG. 168.—Propeller Fan.

fan. The coordinates of these curves are percentages of the performance at rated capacity and refer to any size of the tested type. The rated capacity is that determined at the maximum static efficiency, the point at which the product of static pressure and volume per applied horsepower is a maximum. If the dis

charge pipe on the backward-curved blade fan is closed completely it is found that the static pressure produced is 170% of that obtained when the fan is delivering the cfm of the best static efficiency, called the rated cfm. As the outlet pipe is opened to increase the discharge, there is a slight rise in pressure to a point at which the discharge is $17\frac{1}{2}\%$ of the rated capacity, after which the pressure falls, and when the pipe is opened up so that the fan is discharging 145% of its rated capacity, the static pressure reaches zero. For the fan with forward-curved blades, the pressure curve, starting closed with a pressure at 125% of static pressure

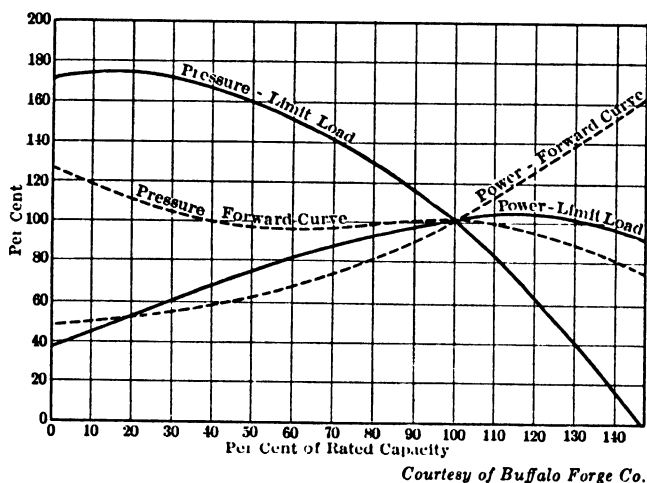


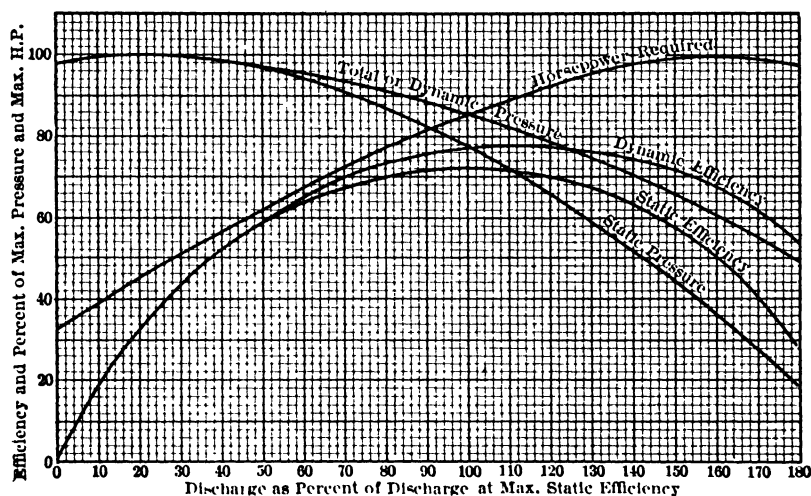
FIG. 169.—Curves of Fan Performance.

at rated discharge, has a slope which changes direction. At a point near capacity discharge the curve slopes in such a direction that, when two fans are in parallel, the loss of discharge by one would continue this loss and give unequal loading. With the rising characteristic of the pressure curve of the Limit Load Conoidal fan throughout its length as load is decreased (except near the zero discharge) any tendency to rob a fan of its discharge would result in an increase of static pressure for that fan which would force the air from it to equalize the discharge from two fans in parallel.

The power curves for these fans are important. With the backward-curved blades the power at zero discharge (caused by churning action) gradually increases with the discharge until a maximum value is reached, after which the power decreases because the pressure decreases at a faster rate than the rate of increase of air discharge. There is no danger of burning out the motor if a break occurs in the duct system, or if free dis-

charge were permitted by any means. This point of maximum power is the basis for the name used by the Buffalo Forge Company for this fan. The power curve of the fan with forward-curved blades increases with discharge to values much greater than that at rating.

In the operation of any one type of fan at a given percentage of the rated discharge at best efficiency for different speeds of rotation at these points, there is found to be a fixed relation between the static pressure, h_s , velocity pressure, h_v , and tip speed (peripheral speed) pressure, h_p , at any speed and for any size of fan of this type, Fig. 170. These are



Courtesy of Buffalo Forge Co.

Fig. 170.—Curves of Fan Performance.

measured in inches of water. The curve of total or dynamic efficiency is obtained from the test data by

$$\frac{144 \times 0.577 (h_s + h_v) \text{ cfm}}{16 \times 33\,000 \times \text{hp}} = \eta_t \quad (9-11)$$

The static efficiency is given by

$$\frac{1.57 \times 10^{-4} h_s \text{ cfm}}{\text{hp}} = \eta_s \quad (9-12)$$

hp = horsepower input to fan.

If now the rotor speed of a given fan is increased, or a fan of the same type but of large diameter is operated at the first shaft speed of the small fan, the peripheral speeds will be greater than before, and the

velocity of discharge will be greater than the original velocity of discharge; but if the delivery in each case is fixed at the same percentage point of the rated delivery for these speeds, the relations between the velocity heads of discharge, peripheral speed or static head will remain the same. Thus, if the fan is delivering the same fraction of its rated discharge at any speed of turning, the velocity of discharge will bear a fixed relation to the peripheral speed. The peripheral speed, V_p , being πDN , the velocity of discharge, V_d , will vary as DN , where D is the outside diameter of the wheel in feet, and N represents the revolutions per minute.

For points of delivery of the same percentage of rated delivery, the following relations hold:

$$V_d = k_1 V_p = k_1 \pi DN = k_2 DN \quad (9-13)$$

For the velocity pressures

$$h_v = x \left(\frac{V_d}{4005} \right)^2 = k_3 x V_d^2 = k_4 N^2 D^2 x \quad (9-14)$$

(x = relative air density)

V_d = discharge velocity in feet per minute.

Also

$$V_d = k_5 \left(\frac{h_v}{x} \right)^{1/2} \quad (9-15)$$

The static pressure, h_s , also bears a fixed relation to the discharge velocity pressure at a given point of discharge, Fig. 170, so that

$$h_s = k_6 h_v = k_7 N^2 D^2 x \quad (9-16)$$

$$V_d = k_8 \left(\frac{h_s}{x} \right)^{1/2} \quad (9-17)$$

$$h_v = 1/k_6 h_s = k_8 h_s \quad (9-18)$$

Now, the areas of geometrically similar (homologous) fans vary as D^2 so that

$$\text{cfm} = V_d A = k_9 V_d D^2 = k_{10} \left(\frac{h_s}{x} \right)^{1/2} D^2 = k_{11} N D^3 \quad (9-19)$$

The work done per minute is equal to the volume of air in cubic feet per minute multiplied by the pressure increase in pounds per square foot. This pressure is the dynamic pressure of discharge, if air is drawn from a place of atmospheric pressure, which bears a fixed relation to the

static pressure for all points at a fixed percentage of the rated discharge for a given rpm. Hence,

$$\text{Air hp} = \frac{144 \times 0.577}{16 \times 33\,000} h_d \text{ cfm}$$

$$\text{hp} = k_{12} h_d \text{ cfm} = k_{13} \frac{(\text{cfm})^3}{D^4} x =$$

$$k_{14} V_d^2 x \text{ cfm} = k_{15} V_d^3 D^2 x = k_{16} \frac{h_s^{3/4}}{x^{1/2}} D^2 = k_{17} (x D^5 N^3) \quad (9-20)$$

The horsepower required to drive the fan will vary in the same manner as the efficiency at a fixed percentage discharge is a fixed quantity.

From these relations it is seen that for any one of the homologous wheels operating at a given percentage of rated discharge

$$\text{Required hp varies as } x N^3 D^5, \frac{h_s^{3/4} D^2}{x^{1/2}}, x V_d^3 D^2, V_d^2 x \text{ cfm}, \frac{(\text{cfm})^3}{D^4} x \text{ or } h_s \text{ cfm} \quad (9-21)$$

$$\text{cfm varies as } N D^3, \left(\frac{h_s}{x}\right)^{1/4} D^2 \text{ or } V_d D^2 \quad (9-22)$$

$$h_s \text{ varies as } D^2 N^2 x \text{ or } \frac{(\text{cfm})^2}{D^4} x \text{ or } V_d^2 x \quad (9-23)$$

$$N \text{ varies as } \frac{V_d}{D}, \frac{1}{D} \left(\frac{h_s}{x}\right)^{1/4}, \frac{\text{cfm}}{D^3} \text{ or } \left(\frac{\text{hp}}{x}\right)^{1/4} \frac{1}{D^{3/4}} \quad (9-24)$$

$$V_d \text{ varies as } N D, \left(\frac{h_s}{x}\right)^{1/4}, \frac{\text{cfm}}{D^2} \text{ or } \left(\frac{\text{hp}}{x}\right)^{1/4} \frac{1}{D^{3/2}} \quad (9-25)$$

x = relative density (to air at 70° F and 29.92 in.)

$$\begin{aligned} &= \frac{\text{Weight of 1 cu ft of given pressure and temperature}}{\text{Weight of 1 cu ft of air at 70° F and 29.92 in.}} \quad (9-26) \\ &= \frac{491.6 \text{ Bar}}{29.92 T} \end{aligned}$$

These may be stated as the laws of fan performances as given by the Buffalo Forge Company in Fan Engineering.

1. For a given size fan, piping system and air density (D , and x fixed and h_s varying as V^2).

(a) When speed varies (N or V_d):

Capacity (cfm) varies directly as N or V_d

Pressure (h_s , h_v or h_d) varies as N^2 or V_d^2

Horsepower (hp) varies as N^3 or V_d^3 .

(b) When pressure varies:

Capacity (cfm) varies as $h^{1/2}$ (h_d, h_s, h_v)

Horsepower varies as $h^{3/2}$ (h_d, h_s, h_v).

2. For constant pressure, density and point of rating.

Capacity or horsepower varies as D^2

Speed (N) varies inversely as D .

✓ 3. For variation of speed and fan size with fixed density.

Capacity varies as ND^3 or $V_d D^2$

Pressure varies as $N^2 D^2$ or V_d^2

Horsepower varies as $N^3 D^5$ or $V_d^3 D^2$

Horsepower varies as (cfm) h .

4. For constant pressure (h) on a given size of fan.

When x varies:

Speed (V) or (N), cfm and hp vary inversely as square root of density or directly as square root of absolute temperature.

5. For constant capacity (cfm), speed (V_d or N) and size.

Horsepower varies as x , or directly as barometer and inversely as absolute temperature.

6. For constant amount by weight [(cfm) x] and fixed size.

h_s varies inversely as density, and as cfm now varies with h_s as well as with N and V_d ; all four h_s , cfm, N and V_d vary inversely with density and barometric pressure but directly with absolute temperature.

Horsepower varies inversely with the square of density or the square of barometric pressure or directly as the square of absolute temperature.

7. For variation of pressure, h_s , and temperature (density) with fixed size.

Capacity and speed vary as the square root of the quotient of pressure divided by density, which is proportional to the square root of the product of static pressure and absolute temperature.

The horsepower varies as the square root of the product of the cube of the pressure and absolute temperature.

Specific Speed. If D is eliminated from the expressions (9-19) and (9-16),

$$\text{cfm} = k_{10} \left(\frac{h_s}{x} \right)^{1/2} D^2$$

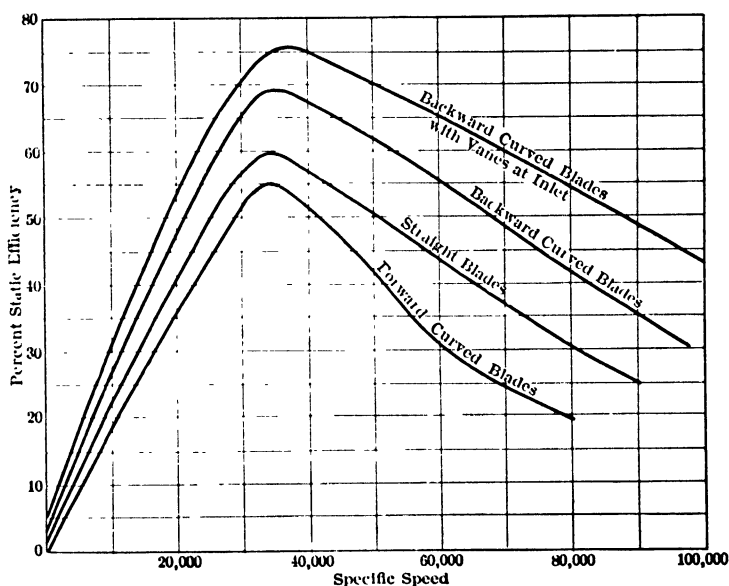
and

$$h_s = k_7 D^2 N^2 x,$$

there results

$$\begin{aligned} \sqrt{k_{10}/k_7} &= \text{Constant} = N \sqrt{\frac{\text{cfm } x^{3/4}}{h_s^{3/4}}} \\ &= \frac{N \sqrt{\text{cfm}}}{h_s^{3/4}} \quad (\text{when } x = 1) = N_s = \text{Specific speed (9-27)} \end{aligned}$$

This quantity, called the specific speed, is the speed of a homologous wheel which would discharge 1 cu ft of standard air per minute with a



From Fan Engineering, Buffalo Forge Co.

FIG. 171.—Curves of Specific Speed.

static pressure of 1 in. This characteristic is the same for fans of any size if they are of the same shape and proportions (homologous). The efficiency-specific speed curves for the several fans of different types are given in Fig. 171. If such a curve is known for any given type of fan, then by selecting the specific speed of the fan near best efficiency the actual rpm for this type of fan at a given cfm and h_s may be found, and, by reference to the complete set of tables of the performances of fans of this type the nearest size to these values of volume, static pressure and speed can be selected.

The fan manufacturers publish tables of performance of each size of

fan with standard 70° air at different static pressures for given velocities of discharge (velocity pressures). These are similar to the partial tables of performances of one size of each of the two fans shown in Figs. 164 and 165, which are given in Tables A-20 and A-21. In these tables, points of the same discharge velocity (velocity pressure) and same static pressure for any one of the fans represent points of the same relation between these two pressures, and consequently they are points at which the discharge is at the same percentage of rated discharge, and by fan law 3, the capacity and horsepower (with the same V_d) vary as the square of a linear dimension (D^2) and the speed N varies inversely as D . Although a table for one size only is given, the performance of other sizes for the same pressure points can be readily found by using the fan laws. In actual office work a complete set of tables gives the data for each size.

These points vary on each side of the point of highest static efficiency which for the limit load fan occurs approximately at the point where $h_v/h_s = 0.08$ or $h_v/h_d = 0.074$. For certain fans with forward-curved blades the most efficient point is at approximately $h_v/h_s = 0.40$ or $h_v/h_d = 0.29$ and others have ratios similar to those first mentioned. At the point of highest total efficiency of the first fan the ratio h_v/h_s is 0.12 approximately.

The sizes of the wheels of the Silent Vane fan and of its parts are given in the Table A-22. The designation number of the Limit Load Conoidal fan is $\frac{1}{8}$ of the number of inches of the diameter of the rotor or wheel. From this number, the diameters of wheels can be found and the other dimensions of the fans, as given in the figure for size 10 attached to Table A-20, can be determined from the dimensions on this figure by realizing that these dimensions bear the same relations to the wheel diameters for all homologous wheels.

The Buffalo Forge Company recommends discharge velocities 800 to 1000 fpm for supply fans of churches and auditoriums, 1000 to 1400 fpm for theatres and classrooms and 1200 to 1600 fpm for restaurants, hotels, stores and offices. The last value is the highest which is recommended. Exhaust fans of such systems, when used, may have discharge velocities 200 fpm greater than the values mentioned for supply fans.

Since the fans are tabulated for 70° F air at 29.92 in. pressure it is necessary to reduce all data to 70° F before selecting the fan.

The school building of this chapter requires 26,460 cfm of 70° air, and the total pressure is 1.0297 in. at 73° F. By the laws of fan performance, the pressures at different densities for a fixed weight of air will vary inversely with the density so that the pressure at 70° F will be 1.0297×0.996 , or 1.025 in. For this school the velocity of discharge will be between 1000 fpm and 1400 fpm and the Limit Load

Conoidal fan No. 10 will deliver 26,460 cfm at a velocity slightly over 1300 fpm. If this is thought too great for a duct velocity of 1000 fpm, a larger size may be selected from tables, if available. With only the one table given in this book, data for larger fans must be computed. At a given pressure and density the capacity varies as D^3 . This means that the number of the fan to deliver 26,460 cfm at a desired discharge of 1000 fpm is

$$10 \times \left(\frac{26\,460}{20\,208} \right)^{1/3} = 11.4$$

(20,208 cfm discharge from No. 10 at 1000 fpm)

The No. 11 fan will be too small and No. 12 will be too large. Therefore No. 11 will be used but at a slightly higher discharge than 1000 fpm (discharge varies as speed for fixed h_s).

Capacity of No. 11 at 1000 fpm = $20,208 \times \left(\frac{11}{10}\right)^3 = 24,440$ cfm

Velocity for 26,460 = $1000 \times \frac{26\,460}{24\,440} = 1080$ fpm.

Velocity pressure of 70° air at 29.92 = $0.0622 \times \left(\frac{1080}{1000}\right)^2 = 0.072$ in.

Static pressure = $1.025 - 0.072$ in. = 0.953 in.

By interpolation from the table for No. 10 wheel with S.P. = 0.95 in. and 1080 fpm.

$$\text{rpm} = 289$$

$$\text{hp} = 4.43$$

For No. 11 (constant pressure, density and point of rating)

$$\text{rpm} = 289 \times \frac{11}{10} = 262 \text{ rpm}$$

$$\text{hp} = 4.43 \times \left(\frac{11}{10}\right)^3 = 5.37 \text{ hp}$$

(N varies as $1/D$, hp varies as D^3)

The power, pressure and speed will now be changed for the same weight of air at 73° by the laws of fan performance.

$$\text{rpm} = \frac{262}{0.996} = 263 \text{ rpm} \left(\text{varies as } \frac{1}{x} \right)$$

$$\text{cfm} = \frac{26\,460}{0.996} = 26,550 \text{ cfm} \left(\text{varies as } \frac{1}{x} \right)$$

$$\text{hp} = \frac{5.37}{(0.996)^3} = 5.40 \text{ hp} \left(\text{varies as } \left(\frac{1}{x}\right)^3 \right)$$

This fan selected will be a horizontal fan and will have the following over-all approximate dimensions:

$$\text{Height } 108.62 \times \frac{11}{10} = 119 \text{ in.}$$

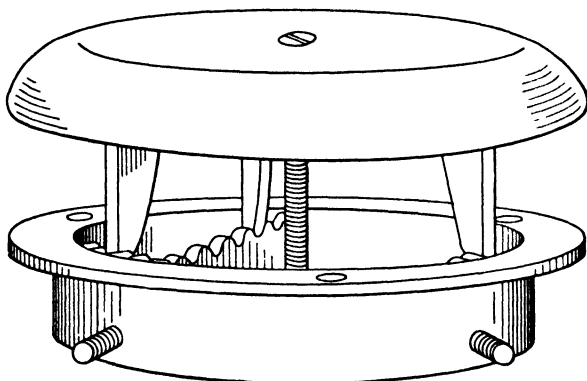
$$\text{Depth } 73.12 \times \frac{11}{10} = 81 \text{ in.}$$

$$\text{Width } 98.75 \times \frac{11}{10} = 109 \text{ in.}$$

Other Plenum Details. In auditoriums, as shown in Fig. 4, the warm air is usually introduced from or withdrawn by mushroom delivery

heads, Fig. 172, placed beneath seats with a low velocity of movement. When used for supply these heads are fed from ducts, or the whole space beneath the floor is a large plenum chamber. In churches, air may be introduced from such heads or from the ends or sides of pews. When cool air is introduced in summer time better results are obtained by bringing the conditioned air from the top of the room and taking the exhaust air through these heads set in the floor.

As an upward flow of air is not as effective in heating before occupancy, the duct system may be arranged with a large revolving cylinder valve so that, by turning this through a quarter turn, air is delivered to the top of the auditorium by the return duct, while the ducts leading to the heads in the floor become suction lines. After warming, this valve is rotated to the upfeed position for use with occupants in the room.



Courtesy of Knowles Mushroom Ventilator Co.

FIG. 172.—Mushroom Discharge Head.

The ozonators, if used, are placed in the discharge duct from the fan. Some have had excellent experience with such treatment (St. Louis school system), but others believe that permissible concentrations can be of little value.

Window Frosting. If the moisture content is too great in the cold season some of this moisture will condense on the windows; to prevent this, double sash are needed. Willard and his associates* have shown that the inside surface temperature of single glass in a 70° room varies on a straight line from 13° F in -10° F weather to 70° in 70° F weather, while with double windows this straight-line variation of temperature of the inner glass on the room side varies from 47° to 70° F. Moisture will condense with the following relative humidities in the 70° rooms with

* Bulletins 141, 266, Engineering Experiment Station, University of Illinois.

single glass: 12% at -10°F , 18% at 0°F , 30% at 20°F , 49% at 40°F . With double windows these relative humidities become 45%, 49%, 60% and 75%. The reason for double windows with 40 or 50% relative humidity is quite evident.

PROBLEMS

1. A building requires 10,000 cfm of standard air at 70°F from outside air of 30°F dry bulb with 20°F wet bulb and 5000 cfm of standard 70°F air from recirculation at 68°F and 60 grains of moisture per pound of dry air. What are the temperature and moisture content of the resultant mixture?

2. If air at 68°F and 50 grains of moisture is to be delivered from a washer at its dew point, how much heat must be added between the washer and the point at which the condition is that given above? What Flexitube heater would be installed to do this for 15,000 cfm of equivalent air of 70°F ?

3. What should be the condition of air of 20 grains of moisture per pound of dry air so that on passing through a washer with water at 49°F it will be discharged in a saturated condition at 49°F ?

4. What size Flexitube heater would be required to heat 15,000 cfm of 70°F standard air of 20 grains of moisture per pound of dry air from 30°F to 60°F ?

5. Find the number of square feet of surface and the size of the unit to heat 30,000 cfm of standard air from 30°F with 10 grains of moisture per pound of air to 110°F by Aerofin heaters, by Vento heaters and by blast pipe coil heaters. Find the friction drop in each.

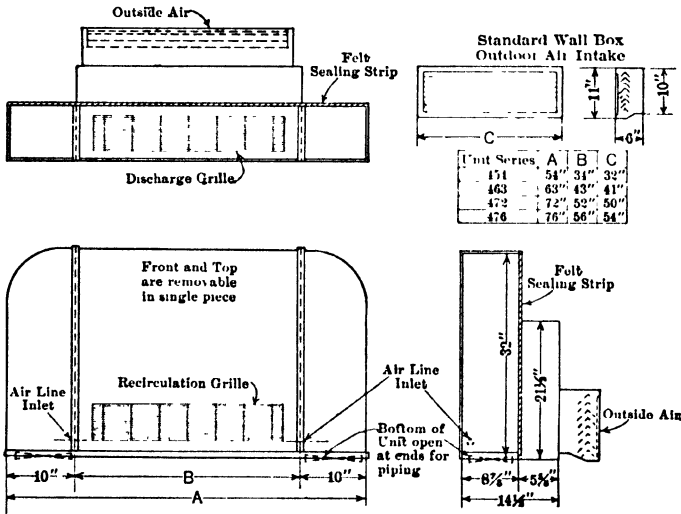
6. A main duct handles 20,000 cfm of 70°F standard air at 90°F . After leaving the heater by a draw-through fan the duct extends 30 ft to the first branch from which 5000 cfm are taken. The second branch of 4000 cfm is taken off at 60 ft from the fan. The next branch at 90 ft from fan removes 4000 cfm, and the main extends to the end of the supply line which is 120 ft from the fan. At this end, six equal branches extend to outlets. The branches are 20 ft long with grilles at the end, and they are attached to the end of the main by elbows with $R/W = 1$. The grilles are attached to the branches by elbows with $R/W = 0.5$. Each branch also contains two elbows with $R/W = 0.5$. Find the sizes of the mains, reducers and branches to be used. Sketch these. Find drop in pressure from fan to room. If there is -0.6 in. of water at fan inlet, what is the total drop in pressure? Select the fan to deliver this air. Find the size of motor required. Find the space requirements of the fan.

7. A room seating 1000 persons is to be maintained at 68° dry bulb and 50% relative humidity. Twenty kilowatts of illuminating power are used. The heat gain from building and sun is 200 Mbb. What quantity of air and what quality of air should be introduced into this room per hour? How much of this should be fresh air?

CHAPTER X

UNIT HEATERS—AIR CONDITIONING

The application of unit heaters to rooms for the supply of heat, the circulation of air and the control of air conditions is gradually increasing. For schoolhouse heating their use is extensive. One of the leading types of unit heaters is that built by John J. Nesbitt and bears the trade name of the Nesbitt syncrétizer. The unit, Fig. 173, consists of a casing con-



Courtesy of John J. Nesbitt, Inc.

FIG. 173.—Nesbitt Syncrétizer Unit Heater.

taining a front inlet for recirculated room air, a back inlet for outside air, a horizontal top outlet for air delivered by unit fans over a Nesbitt extended fin radiator. The figure gives the dimensions for the four sizes which deliver 500 cfm, 750 cfm, 1000 cfm and 1250 cfm of 70° F air respectively.

The radiator receives heat throughout its length at all rates of steam flow, resulting in a uniform temperature of discharge air, by the use of

a valuable unique invention. The steam tubes of the finned radiator are supplied with steam from the supply boxes by two internal tubes each containing a series of small holes distributed over half the length of the main tube, Fig. 174, so sized and placed that steam reaches all parts of the length of the main radiator pipe even at times of small

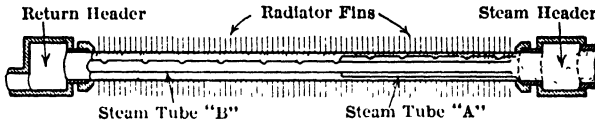
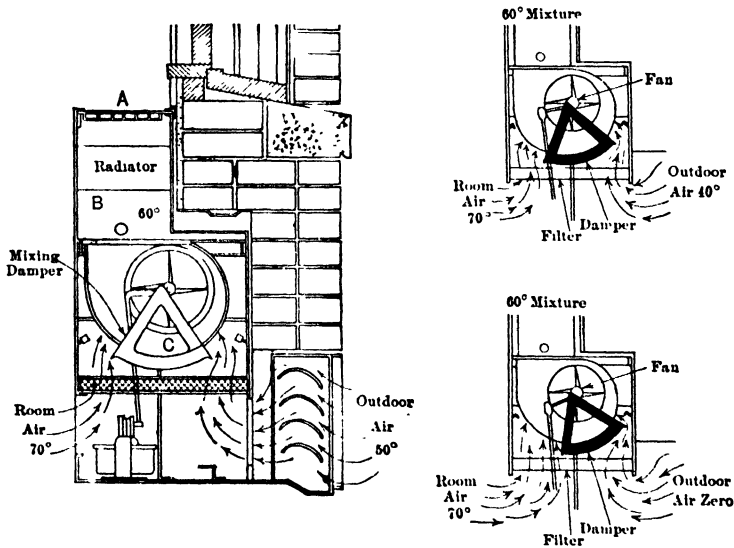


FIG. 174.—Nesbitt Radiator Steam Tubes.

steam flow. Were the steam admitted freely to the ends of plain steam tubes it would be condensed by the fins at the end of the tubes near the supply box at such times, leaving the fins at the opposite end without steam. This would give discharge lanes of hot and cold air. By insur-



Courtesy of John J. Nesbitt, Inc.

FIG. 175.—Section of Nesbitt Synchronizer.

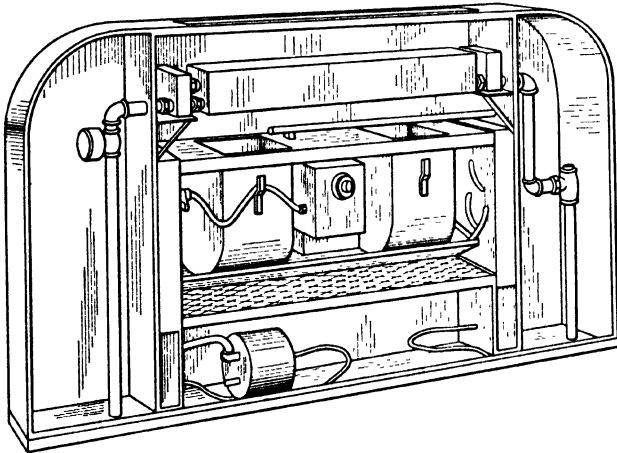
ing a discharge through the small orifices at all parts of the length of the steam tubes in the radiator, the air is uniformly warmed at all times but especially at times of low heat demand.

The felt pads indicated on the back of the unit are also placed beneath it to eliminate noise and make an airtight fit. The units are fastened to the wall and to the floor.

The section, Fig. 175, indicates the way in which the mixing damper,

C, controls the relative amounts of air from two supplies. In the large section, the damper is mixing 50% of 70° recirculated air with 50% of 50° fresh air to give 60° air at entrance to radiator, while the smaller figures show two positions of the damper for the same mixture temperature with outside temperatures of 40° F and 0° F. The mixing damper is moved by an air motor or an electric motor as indicated at the bottom of the section. This motor is under the control of a thermostatic element at *B* or below *A*, as will be seen later. The curved ends of the casing of the unit, Figs. 173–176, contain the valves, pipes and other operating devices.

The general arrangement of a two-fan unit is shown with the front removed in Fig. 176. The recirculated air chamber beneath the filter,



Courtesy of John J. Nesbitt, Inc

FIG. 176.—Nesbitt Syncretizer with Front Plate Removed.

which covers both air supplies, contains the damper motor at the left. The fans are of the double inlet type with the motor between. For greater air quantities four fans are driven by the motor. The perforated brass tube thermostat with an inner bar of stainless steel shown above the fan outlet at the right is known as a leakstat. This device is placed in the position shown for the type O syncretizer, in which the outside air is cut off in the heating-up period but, when the room reaches a point 3 degrees below the desired temperature, the damper begins to function under the control of the leakstat and outside air is admitted to maintain a temperature of 60° F at inlet to the radiator. In the type F device the leakstat is placed above the radiator to control outlet temperature by regulation of steam supply since with this type all the air is taken from

the outside during occupancy of the room. During the heating-up period to 3 degrees below desired air temperature all air is recirculated from the room. In the type A unit (third type) the same method of heating up is used but after the 3 degree temperature limit is reached the damper is moved to admit a minimum quantity of fresh air. This quantity cannot be changed as long as the air leaving the radiator is not above a definite value. When this temperature becomes too high not only is the steam shut off from the radiator but the damper is then opened to admit more outside air. Each one of these types is used to satisfy certain conditions fixed by the desire of the designer or by state laws.

The units may be employed in connection with direct radiators placed in the room for part of the heating (modified split system), or unit heaters alone may be used. In this latter case the heat requirement of the room is fixed and the unit heater selected for the duty.

The following data are given for room 306 of the school discussed in Chapter IX.

Room 306

Heat loss...	24 0 Mbh
Air circulation (at 70° F)	1470 cfm
Bodily heat (sensible)	10 0 Mbh
Sun effect (none on cloudy day)	
Heat demand in warming up...	24.0 Mbh
Heat demand during occupancy without fresh air...	11 6 Mbh
Heat demand for 50% fresh air	
$\frac{1}{2} \times \frac{1470}{55.6} \times 60 \times 70$	55.6 Mbh

The selection of the unit is made by obtaining the equivalent direct radiation for the space to be heated as these unit heaters are so rated in the data sheets of the manufacturers.

To supply 69.6 Mbh, 290 sq ft of equivalent direct radiation (steam) would be needed. Hence, two 463-3A syncrétizers, each circulating 750 cfm of standard air of which 50% is 0° F air and each containing 172 sq ft equivalent direct radiation, would be selected from one of these data sheets, Table A-24.

Data sheets have been prepared by the manufacturer for performances with various amounts of outdoor air for the type A syncrétizers and for single amounts for type F and for type O. On account of different heat losses from rooms of the same air requirements the radiators must have one of four different fin areas per foot of length. These data sheets give all information that is necessary for selection and installation.

The unit heaters are usually placed beneath windows on the outside wall and near the center of the length of the wall, distributing air at

750 fpm against the ceiling, where it spreads, and gradually falls so that the recirculated air returns to the inlet grille at velocities through the room of not more than 40 fpm.

Outlet duct systems should always be used to remove an amount of air equal to the fresh air admitted. Air is taken through the wardrobe room to the corridor from which the flues lead to the attic and thence to the ventilator heads on the roof or to an exhaust fan. Ventilator heads, Fig. 177, are activated by wind velocities across their tops which produce suction and remove air from the vertical shafts under the con-

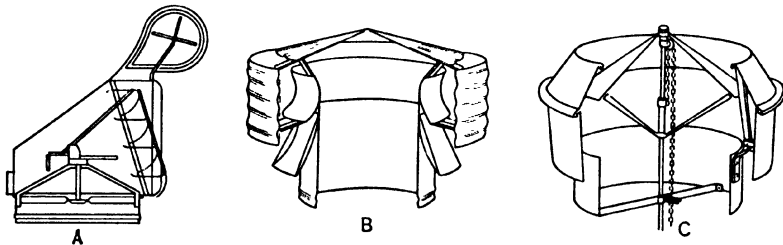


FIG. 177.—Ventilator.

trol of dampers. The construction of the parts is such that rain cannot enter the sheet metal flue. The various types of ventilator heads are rated by their manufacturers so that the designer may select a size suitable for the amount of outlet air. In general, the velocity in the flue may be taken as 200 fpm in still air, 450 fpm at 5 mph, and 950 fpm at 15 mph.

Tests have been made in schools with closed outlet ducts, windows and doors, and under these conditions it was found that the air discharge from the unit heaters was decreased by about 10%, an amount which could be cared for by speeding up the fans and using more energy in the motor. Whether this increased annual energy cost would be paid for by the saving in annual cost from the elimination of the exhaust ducts is not as much the question as the slightly higher pressure which must be carried to force this air through cracks in windows, walls and doorways. The increase of speed also means a possibility of noise.

Unit Air Conditioners. For summer air conditioning and winter heating such units as shown in Fig. 19 are used, and to illustrate they will be applied to a room which is to be air-conditioned with an outside temperature of 90° F with an 80° wet bulb. The 40 × 60 ft room has 300 sq ft of wall and 140 sq ft of glass on the south and 450 sq ft of wall and 210 sq ft of glass area on the east and on the west sides. The value of U for the wall is 0.18. The room is above one with its ceiling at 85° F and below one with the floor at 80° F. The value of U for the floor and

ceiling is 0.26. The capacity of the room is 26,000 cu ft. There are 60 people at light work in the room. The lights and motors use 14 kw of power. It is desired to hold this room within the summer comfort zone at 77° dry bulb with 67° wet bulb, an effective temperature of 73° F with some recirculation through the unit conditioner and an amount of fresh air equal to 10 cfm per occupant. The temperature at the floor will be taken as 75° F, and that at the ceiling 79° F. The fresh air will amount to 36,000 cfh and will be taken to the conditioner unit by a duct to prevent its mixture with the room air. In solving this problem, it is to be noted that the method used is applicable to any conditioning problem for offices, auditoriums or factories as the elements of design are included.

The moisture per pound of outdoor air at 90°–80° is 139 grains per pound of air as found from Fig. 24. The enthalpy is 43.7 Btu per lb of air. The enthalpy per pound of room air at 77°–67° is 31.65 Btu per lb of air with 83 grains of moisture per pound of air. The specific volume of outside air is 14.39 cu ft per lb of air, and that of the room is 13.78 cu ft per lb.

HEAT GAIN PER HOUR

Heat through walls	2 810 Btu
$1200 \times 0.18 \times (90-77)$	
Heat through glass	8 240
$560 \times 1.13 (90-77)$	
Heat through floor	6 240
$2400 \times 0.26 (85-75)$	
Heat through ceiling	624
$2400 \times 0.26 (80-79)$	
Sun effect	28 040
Roof	000
East or west wall	1 860
$450 \times 0.18 \times 23$	
East or west windows	27 500
$210 \times 196 \times 0.68$ (shades, no awnings)	
South wall	540
$300 \times 0.18 \times 10$	
South windows	7 680
$140 \times 81 \times 0.68$ (shades, no awnings)	
Use west windows and south wall on account of time lag of wall	
Heat with entering air	30 300
$\frac{36\ 000}{14.39}$ (43.7 – 31.65)	
(above room condition at which it escapes)	
Heat from occupants (sensible and latent)	31 500
60×525 (mean of 380 and 670)	
Heat from electrical energy	47 740
14×3410	
Total heat to be removed per hour by coils	155 494 Btu

The refrigeration needed is expressed in tons of refrigeration. The ton of refrigeration is by definition the equivalent of the removal of 200 Btu per min, which is the rate at which the melting of a ton of ice in 24 hr would extract heat.* The refrigeration for the unit conditioners in this room would be 12.9 tons, and this would be the peak which would occur near 4:00 P.M.

The large load could be reduced by awnings although the largest amount of heat to be removed is produced by the occupants and the electrical energy.

The moisture added per hour from perspiration, taken from Fig. 28, is 1850 grains per person as light work is done. This total moisture of 111,000 grains per hour will have to be removed by condensation, but no heat allowance is made in the room for the heat of this moisture (called latent heat from persons) although it must be included in heat removed by the conditioner. The latent heat does not affect the change in temperature of the cooled air delivered to the room.

The moisture in the room air is maintained at 83 grains per pound of air; with outside air at 139 grains per pound of air, the moisture to be removed in the conditioner from the fresh air is 141,000 grains per hour,

$$\frac{36\,000}{14\,39} \times (139 - 83) = 141,000$$

The total moisture to be removed is 252,000 grains or 36.0 lb per hour.

The fresh air will be delivered to the unit heater to prevent the mixture of this hot air with the cool air of the room. The air will be delivered from the conditioner at a dew point of 83 grains per pound of air (62° F) so as to provide the desired moisture content, and the amount of air circulated will be fixed so that this amount of air at the dew point will abstract from the room air on mixing with it the amount of sensible heat given to the room from all sources except the fresh air which will be cared for by the coils of the conditioner and the latent heat from the occupants which has no effect on temperature change in room. If the temperature of the dew point is thought to be too low for delivery to the room,† the air delivered must be discharged over a heat interchanger with the return air on the opposite side of the surface, although this will increase the amount of air to be handled by the fans. When the dew points are 20° below room air, the dried air must be heated before delivery.

The latent heat from the occupants is equal to 275×60 or 16,500 Btu per hr as shown by Fig. 28.

The dew point at 83 grains per pound is 62° F; the enthalpy for this is 27.88 Btu per pound of air, and its volume is 13.40 cu ft per lb of air. The weight of air to be delivered into the room air to maintain the enthalpy 31.56 Btu per lb of dry air will be

$$\frac{155\,494 - 30\,300 - 16\,500}{31.65 - 27.88} = 28,800 \text{ lb per hr}$$

The volume of this air from the unit is 396 000 cfh or 6600 cfm and equals 15 changes per hour. This air entered the cooler at a temperature 75.7° F dry bulb as shown in the next problem.

* This is the definition of the ton of refrigeration, although the melting of a ton of ice in 24 hr is equal to 199.7 Btu abstracted per minute.

† A difference of 18° F from room temperature may be used if inlet air is distributed at high velocity from grilles near ceiling.

This amount of air receives only the moisture from perspiration, as the extra moisture from the fresh air is removed in the unit conditioner. The increase of moisture by perspiration or "pick-up" is 3.85 grains per pound of air

$$\frac{111\,000}{28\,800} = 3.85$$

This will be scarcely appreciable.

This circulation is four times as much as the usual amount of 1800 cfm per person, but if this peak load must be carried and the air of the room maintained at 77° F dry with 67° F wet bulb with the various sources of heat, this quantity of air circulation is necessary.

If a heat interchanger were built to deliver the circulation air at 70° with 83 grains per pound (with an enthalpy of 29.5 Btu per lb) the circulated air would be increased to 15,000 cfm.

The data sheets of the Buffalo Forge Company for the floor units or the flat suspended units with steel or copper tubes give this capacity with No. 354 F.S.S., Fig. 19, the flat suspended cooler with steel pipes, Table A-25. This delivers 3200 cfm at 850 rpm with a heat-abstracting capacity of 2340 Btu per hr per deg difference between entering air and leaving refrigerant. Since the total capacity needed is 6600 cfm of air and 155 494 Btu per hr, two units with a 33° temperature differential would be needed for this room. This would mean a brine refrigerant at 42.7° at entrance and 59° F (75.7 - 33 × 0.51) at the point of air discharge. Each unit would require 40 gpm of brine, and the power taken by the fans would be $\frac{1}{2}$ hp. The space requirements would be 85 × 50 × 18 in.

Aerofin Continuous Water Coils. To compute the Aerofin coils for this service the inlet air condition and the outlet condition must be known. The data follow:

36,000 ÷ 14.3 or 2510 lb per hr of 90° dry-bulb air with 139 grains of moisture and enthalpy of 43.7 Btu per lb of dry air.

28,800 lb of air delivered at 62° F wet bulb (dew point) with 83 grains of moisture and enthalpy of 27.88 Btu per lb.

26,290 lb of return air with 31.65 Btu of enthalpy but with 86.85 grains of moisture.

The entering air mixture of return air and fresh air will have an enthalpy of 32.7 Btu and 91.3 grains of moisture per pound of dry air. The wet bulb is at 68.3° F and the dry bulb at 75.7° F.

If water at 50° F is available, a line from 56° F saturation (mean of 50° and 62°) to the point 75.7° F - 68.3° F will intersect the 62° wet-bulb line at dry-bulb line 65° F. The cooling water will be heated to 65° F by the coil for a 15° rise.

$$\Delta t_2 = 65^\circ - 50^\circ = 15^\circ \text{ F}$$

$$\Delta t_1 = 75.7^\circ - 65^\circ = 10.7^\circ \text{ F}$$

$$\text{Mean } \Delta t = \frac{15.0 - 10.7}{2.3 \log \frac{15.0}{10.7}} = \frac{4.3}{2.3 \times 0.146} = 12.8^\circ$$

$$\text{Sensible heat removed} = 28,800 \times 0.24 \times (75.7 - 65) = 74.1 \text{ Mbh}$$

$$\text{Total heat removed} = 28,800 (32.7 - 27.88) = 139.0 \text{ Mbh}$$

Sensible heat ratio = 0.533

$$\text{Volume of air at } 70^\circ - 50 \text{ R.H.} = \frac{28\,800 \times 13.52}{60} = 6500 \text{ 0 cfm}$$

$$\text{gpm} = \frac{139\,000}{500 \times 15} = 18.5$$

$$\text{Water velocity} = \frac{18.5 \times 1.235}{18} = 1.26 \text{ fps}$$

Assume 500 fpm for air

$$\text{Area} = \frac{6500}{500} = 13 \text{ sq ft}$$

Nearest size 12.9 sq ft with 6 ft length ($80\frac{1}{2}$ in. \times 29 in. \times 10 in.)

$$\text{Actual velocity} = \frac{6500}{12.9} = 503 \text{ fpm}$$

Constant from figure of Table A-26 is 134

Sensible heat correction is 0.732 from Table A-27

$$\text{Number of rows of tubes} = \frac{74\,100}{134 \times 0.732 \times 12.9 \times 12.8} = 4.6 \text{ tubes}$$

A unit of five tubes in depth will be used. This will be placed in a duct $80\frac{1}{2}$ in. \times 29 in. and supplied with mixed air from a fan, and air at 65° will be delivered to the room at 77° F dry-bulb temperature. In practice the entering air may be as much as 18° below the room air, and as the difference above is only 12° no heating is necessary.

Air Conditioning by Washers. The air conditioning just described could have been accomplished by a washer in which water below 62° would have been sprayed through the washing chamber the water temperature being fixed by the desired temperature of 62° F for the air at discharge from the washer. The data of the Buffalo Forge Company show that, if one washer were used for 6600 cfm, the cross-sectional area of the air passage at 500 fpm would be 13.0 sq ft, requiring a washer 5 ft $4\frac{1}{2}$ in. wide, 4 ft $1\frac{1}{2}$ in. high and 7 ft $2\frac{1}{8}$ in. long, using 36 gpm of spray water. This spray water would have to remove 155.5 Mhh. If the water in the tank reaches 62° F the entering spray from a refrigerating system would be 8.65° F below this or at 53.35° F.

$$\frac{155.500}{60 \times 36 \times 8.3} = 8.65^\circ \text{ F}$$

The 20 gpm of flooding water on the baffle plates will have to be at the final air temperature so that the washer would be arranged with a pump to take the flooding water from the bottom of the spray chamber, while the spray water would be supplied from a water cooler using some refrigerant.

Dehumidification by Adsorption. The moisture content of the air may be reduced by passing the mixture through sand-like pieces of materials which will condense water vapor into its pores, without any

chemical changes, until these grains become saturated under the conditions of pressure and temperature.

Such materials as silica gel (amorphous silica), calcium chloride and alumina are used. The weight of this moisture adsorption may amount to 8% to 25% of that of the material, and after it is saturated it may be reactivated by heat for further use.

Sprays or sheets of solutions of calcium chloride and of lithium chloride adsorb water vapor in the same way until the solutions become so weak that no further action can take place. Then the solutions must be reactivated through concentrating by heat. The concentration of the solutions, the final relative humidity of the air and its temperature are related by the equilibrium of the partial pressures, so that by carrying a definite concentration a definite relative humidity may be obtained.

When the water vapor is condensed by the solid or liquid adsorbers the latent heat of the vapor becomes sensible, and unless heat is removed during this action the dry-bulb temperature of the air rises as the air becomes drier. The air must then be cooled by passing it over coils cooled by available water supply. The wet-bulb temperature of this very dry air is so low that the dry bulb may be decreased by passing the dry air through a water spray chamber, and even though the water is continuously evaporated the adiabatic change along the wet-bulb temperature line maintains constant total heat and the dry-bulb temperature falls. This method of lowering the dry-bulb temperature is called evaporative cooling.

Refrigeration for Conditioners. The medium low temperatures used for the water in air conditioning has led to the introduction of a refrigeration system in which the cooling of this water is accomplished by the evaporation of part of the water in a flash tank at low pressure.

The vacuum system of refrigeration, the Ross decalorator, of the American Blower Company, is shown in Fig. 178. A low vaporization pressure corresponding to such a temperature as 40° F desired in the conditioning water is maintained in an evaporator or flash chamber by a so-called primary ejector operated by steam jets. A float valve admits water to a spray head in this chamber from the water basin at the bottom of the conditioner, some of the water evaporates to satisfy the atmosphere in the chamber, obtaining the heat for this evaporation from the water and cooling that which falls to the bottom of the flash chamber.

The form of the diverging member of the primary ejector is such that the pressure at its discharge end is sufficiently above that at the suction end that the vapor (steam) from the evaporated water and the steam jets may be condensed in a primary condenser by water at a higher temperature than that in the flash chamber. Two steam jet

secondary ejectors are used to remove any air remaining in the primary condenser after the water vapor has been condensed, and they deliver this air against atmospheric pressure in two stages, through the two-part secondary condensers which condense the steam from these jets in succession.

Since the surface condenser and flash chamber are at pressures lower than the atmosphere, a condensate pump and a chilled water pump must be used to deliver the water from these spaces into the atmosphere.

It must be realized that condensing water should be available to remove the heat from the vapors entering the condensers. This heat

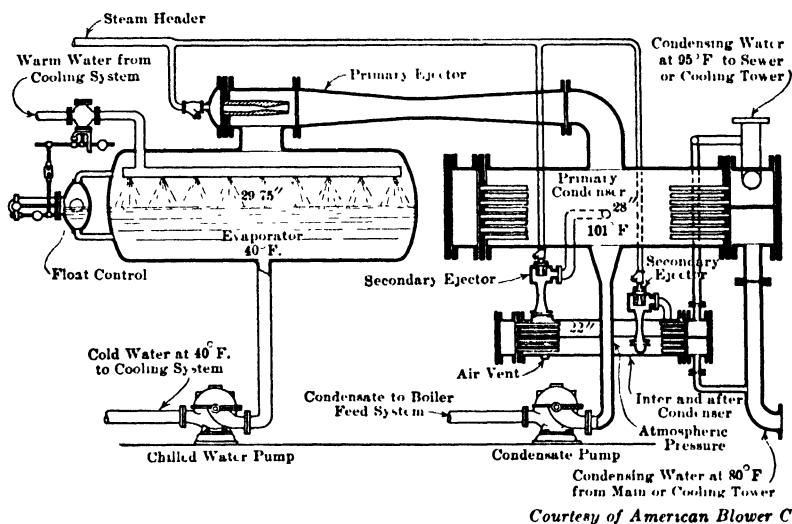


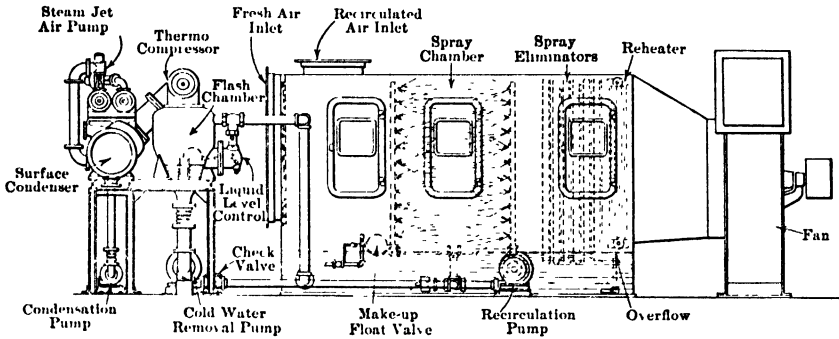
FIG. 178.—Decalorator.

is the sum of that from the evaporated water vapor and that from the steam jets. The condensing water must remove heat equal to the amount of refrigeration plus the amount from the actuating steam. The apparatus has produced refrigeration capacity in water of low temperatures by using available water at a higher temperature. (The ejectors are called thermo-compressors by some manufacturers.)

The vacuum refrigerating system of the Foster Wheeler Corporation is shown in Fig. 179 as applied to an air conditioner. This figure shows the return of water from the basin to the flash chamber and the removal pump which discharges cold water to the spray heads of the conditioner.

If the mechanical compression method of refrigeration is used to obtain the cold water, a compressor, A, Fig. 180, compresses the vapor of a volatile fluid to such a pressure, that, when the vapor enters a con-

denser, C , cooled by available water from D , its vaporation temperature at this pressure will be so much above that of the available water that it will condense, giving up its heat to the condensing water. This liquid from the condensed vapor flows to a receiver, E , and then passes through



Courtesy of Foster-Wheeler Corporation

FIG. 179.—Vacuum Refrigeration System.

a throttling or reducing pressure valve, F , into a set of evaporation coils, B , in which the pressure maintained by the suction of the compressor is so low that the temperature of evaporation of the liquid refrigerant is

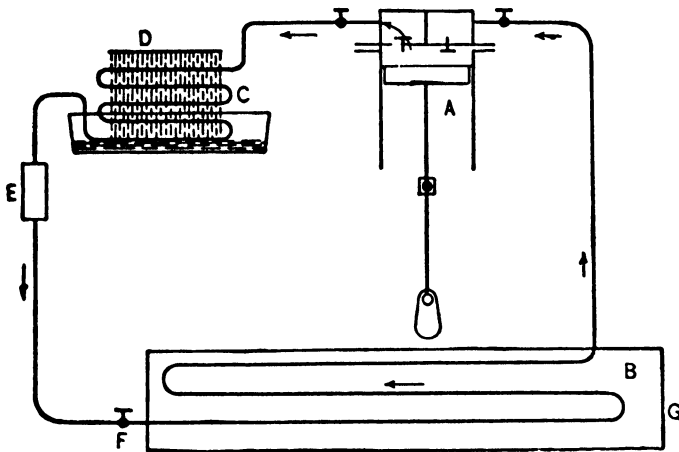
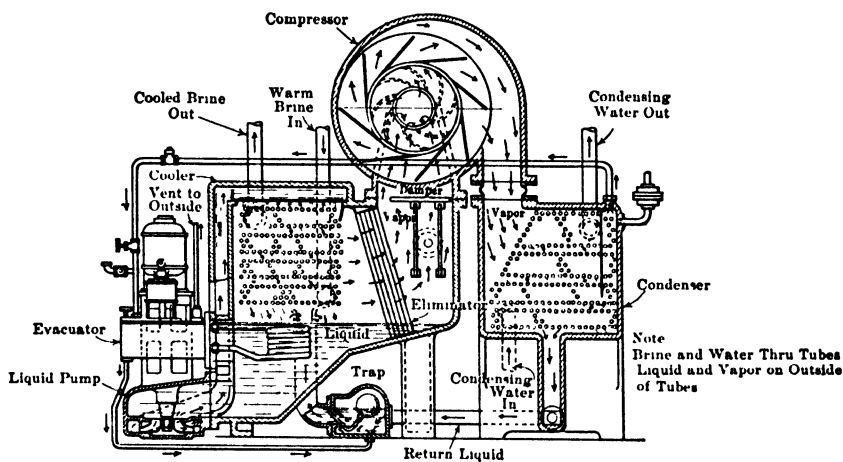


FIG. 180.—Compression Refrigeration System.

below that of the fluid surrounding the coil in the tank, G , and as a result the liquid will evaporate by extracting heat from the substance in G . This substance may be water or brine. The compressor, A , draws this vapor from the evaporation coils for recompression and

maintains the low pressure in coil *B*. The condenser shown is a so-called atmospheric condenser in which the condensing coils are exposed to the atmosphere and water is allowed to fall freely over them. In an enclosed condenser the water is caused to flow over the condensing surface within an enclosing vessel or pipe. In small units the heat is removed by air moving over the condenser tubes by natural circulation or by a blast.

A Carrier centrifugal compression refrigeration machine is shown in Fig. 181. In this machine low pressure is maintained in a chamber on the left by a centrifugal compressor which draws vapor from that space. A volatile liquid in the chamber is discharged over tubes which contain



Courtesy of Carrier Engineering Corporation

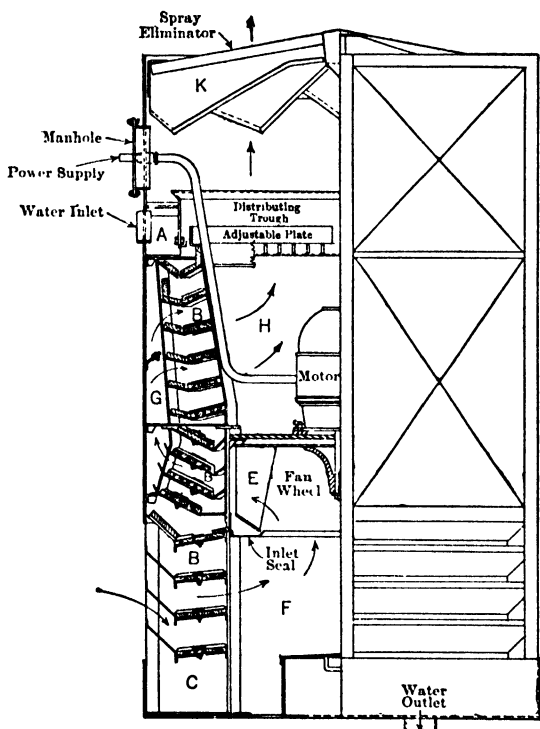
FIG. 181.—Carrier Refrigeration System.

brine or other fluid to be cooled. The liquid at the low pressure produced by the suction of the compressor is evaporated by the heat in the brine or fluid and the vapor from the liquid is delivered from the compressor at a pressure above that of the suction side. The temperature of condensation of this compressed vapor is higher than that of the water which is passed through the condenser tubes on the right, and the vapor is condensed. It falls to a low level and is driven to the liquid tank on the low-pressure side by the higher pressure in this condensing chamber. The liquid is lifted by a centrifugal pump to the spray heads to pass again through the cycle. The volatile fluid used in this machine is known as carrene. Its thermal properties are those desired for the mechanical requirements of this apparatus.

In the compression system, as in the flash system, a condensing

medium (water in the examples described) is needed to remove the heat equal to the refrigeration plus that equal to the energy required to compress the vapor.

The amount of condensing water depends on its temperature range in the condenser, being smaller if this range is greater, but in any event it is appreciable. This water is very expensive when only city supplies are available, and for that reason cooling towers, such as that of Coey, Fig. 182, are installed in which the warmed condensing water is cooled



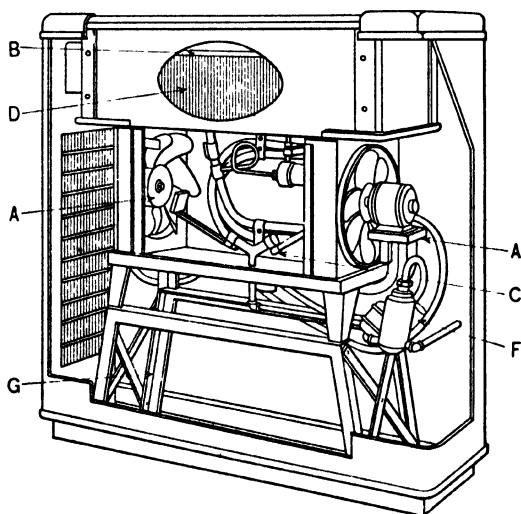
Courtesy of Research Corporation

FIG. 182.—Coey Cooling Tower.

for recirculation and use by bringing it into contact with air circulated through the tower. This water is cooled by warming the circulated air and also by the evaporation of part of the water to saturate this warmed air with water vapor. The heat taken from the water to warm the air and evaporate the required moisture cools the water sufficiently for its return to the condenser. In this way the heat required by the conditioner is taken up by the refrigerated water, and this in turn by the refrigeration system. The refrigeration system then transfers this heat,

and that required for driving the system into the condensing water, to be given up to the air in the cooling tower as sensible heat and heat of vaporization in the added moisture content of the discharge air.

The warmed condensing water is delivered at *A* and falls over perforated diaphragms *B* of the cooling tower, finally reaching the basin *C* in a cooled condition. Air, drawn in from the sides of the tower by the fan *E*, passes through the falling water in three passes at *F* to *G* and *H* and is discharged at *K* in its warmed condition with an increased moisture content. The decrease of the enthalpy per pound of water is equal to the increase of enthalpy of the air and moisture used per pound



Courtesy of Westinghouse Electric and Manufacturing Co

FIG. 183.—Air Conditioning Unit Cabinet.

of water cooled. This gives data by which the necessary quantity of air may be found for a given quantity of water.

At times spray nozzles for the water or cooling ponds may be used for the cooling of this condensing water.

This cost of required condensing water or its cooling must always be considered.

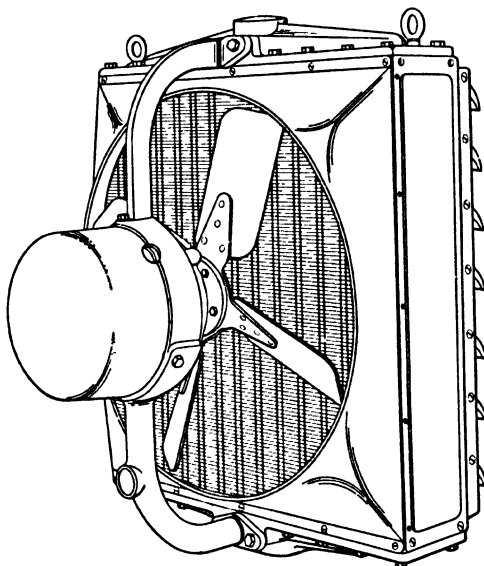
If cold well water is available at temperatures suitable for the conditioner it may be used without refrigeration and the discharge is used as house water in plumbing fixtures. If not too valuable the water may be sent to waste.

It is also possible to use the expansion coil of the refrigeration system

in the air conditioner directly, called direct expansion system, or the expansion coil serves to cool brine which flows in the coils of the conditioner, Fig. 19. A brine coil in the basin or the direct expansion system even may be used to cool the spray water of the washer.

The design and selection of refrigerating apparatus and cooling towers are indicated in textbooks on refrigeration and should be consulted for design procedure.

One of the unit conditioners built to resemble an enclosed radiator for use in a room is that made by the Westinghouse Electric and Manufacturing Company. It is shown in Fig. 183. The metal case, finished to represent mahogany or walnut, covers the device with air inlet grilles at each end and an outlet grille at the top. Two fans, *A*, deliver air over the extended surface radiators, *B*, supplied with steam and hot water in coils for heating; another coil is connected to a supply of cold brine, cold water or a volatile refrigerant for cooling in summer. The moisture removed by the plates *D* from summer air is caught in a drip



Courtesy of Ilg Electric Ventilating Co.

FIG. 184.—Unit Suspended Heater

pan and drained off; for winter humidification two water sprays are placed at *C* under the control of a solenoid valve at *F*. The frame, *G*, is made of several heights to accommodate different desired heights of casing. These units are usually placed in front of wall spaces. Capacities and operating data for steam, refrigerant, electric and water supply as well as drip connection are furnished by the manufacturer.

Fig. 184 illustrates the suspension form of unit heater of the Ilg Company, which is carried from the ceiling or roof of a room and receives steam at the top for the extended surface radiator, the condensate being removed from the bottom by a return trap which is connected with the return system. This unit discharges heated air to the room over the deflectors which may be adjusted. The unit adds the necessary heat to

air at room temperature to equal all or part of the heat losses from the room. The heights of the units are fixed by available head room, but 7 ft is recommended if possible. For a height of 12 ft a high-speed fan is needed, although such height is inadvisable. The heaters should in general discharge along the outside wall for an intermediate story of a building, and the discharge from various units should be directed to aid in forming a definite line of circulation. In a one-story structure or shop building with a monitor the fans deliver downward toward the floor so as to remove the cold air from the roof, and in general they discharge at right angles to the center line of the building. The manufacturer's data sheets give air and heat capacities for units of different sizes as well as suggestions for installation.

Large Unit Heater. The shop building shown in Figs. 5 and 185 is another example of large unit heaters. In this building, 170 ft \times 100 ft, there are 830,000 cu ft of volume, 10,000 sq ft of glass and door area, 15,000 sq ft of wall surface with a transfer coefficient of 0.24, and 18,000 sq ft of roof, with $U = 0.20$. There are 300 men employed, and the tools use an average of 250 kw of electrical power. The leakage from closed pivoted windows on one side with 1500 ft of cracks at 176 cfh per ft with a 15 mph west wind is 264,000 cfh. The wall leakage would amount to 32,000 cfh. The combined leakage of 296,000 cfh of fresh air is more than sufficient for 300 occupants, in accordance with the minimum requirements of the A.S.H.V.E.

The shop is to be held at 60° F in 20° F weather. The temperature at the roof will be taken as 70° F as an average.

Natural Ventilation or Aeration. Should it be desired to introduce more air than that from leakage for ventilation because of the nature of the product, natural air forces may be utilized. The pressure effect on the windward side of the building and the suction effect on the leeward side may be employed for this purpose, as shown in Fig. 185. As the wind strikes the square or inclined faces, it spreads to the edges, leaving the boundaries in such a way that regions of low pressure are formed at the sides and rear of the structure into which air will be drawn from the building. By orienting the building so that many of its low-level windows may be on the side of the prevailing wind, fresh air will be forced in while skylights, monitor windows or even side windows on the leeward side may be used for exhausting vitiated air. Never should windows on both sides of the upper structure be open at the same time, as the induction effect of this flow would not be so great as the effect from a low-level inlet and high-level outlet. The monitor or skylight should be perpendicular to the prevailing wind, as well as the side with ventilating windows, although there is some slight aspiration effect from

the wind when surfaces are parallel to its direction. An attempt should be made to balance the areas of the inlet and outlet openings so that a definite flow may be obtained. By using windows at upper levels for outlet the chimney effect of the hot air of the building between inlet and outlet at different elevations becomes a circulating force in addition to that set up by the wind forces, but there should be no direct path

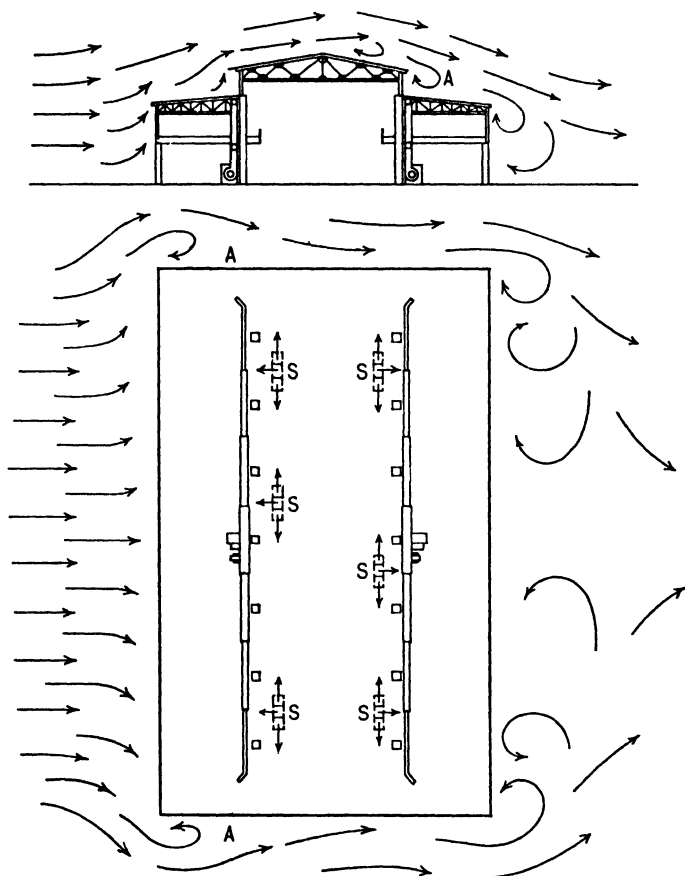


FIG. 185.—Heating and Ventilating of Shop.

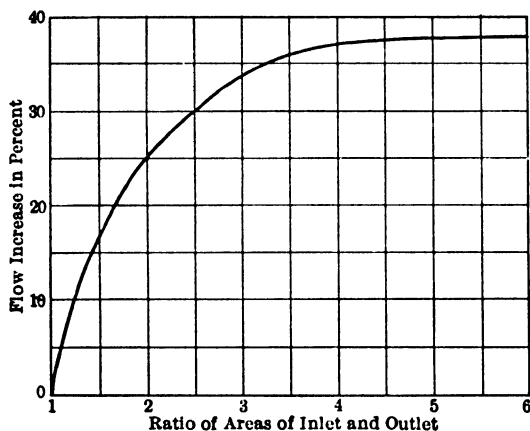
from the supply to the exhaust windows as distribution of the fresh air is needed.

W. C. Randall has reported* on experiments made on actual buildings and models of buildings. In these tests the region of low pressure

* The Aeration of Industrial Buildings, Trans. A.S.H.V.E., Vol. 34, 1928; Vol. 37, 1931.

on the sides parallel to the wind extended back from the windward face as indicated in Fig. 185. Randall states that the length of the jump *A* at the sides or at the flat top of a building is independent of the velocity but is a function of the square root of the face area of the windward side of the building. It increases with the frontal width until the width is about 9 times the height. Although the extent of side region of low pressure is independent of velocity the suction force or drop in pressure varies with the square of the velocity.

The velocity of air entering the windows on the windward side of the flat-roof buildings is taken by Randall to be 50% of the wind velocity for perpendicular flow and 30% when the wind blows at an angle; when combined with monitors for outlet, these quantities will increase to 60% and 35%. If the building is long in the direction of wind movement, the aspiration effect from the side windows increases the inflow velocity



From *Trans. A.S.V.E.*, Vol. 37, 1931, by Permission

FIG. 186.—Effect of Ratio of Window Areas.

from 50% to 64% when the length is twice the width and to 67.5% when the length is three times the width. (With monitors, 60% is increased to 74% and 80% in these cases.) Randall points out that there is usually a prevailing wind direction at each part of the United States and the window arrangements of a building (fenestration) should be oriented to suit the wind direction.

When the amount of air needed for heat removal with a given temperature drop, or that used for some particular condition, is determined, the areas of openings may be found from the required volume and the wind velocity (1 mile per hour is equal to 88 fpm). To obtain the velocities given above, the area for outlet should equal that of inlet. If the outlet area is increased the inlet area may be decreased, and if it is necessary to decrease the outlet area then the inlet area must be increased. The effect of the inequality of these areas is given by Randall in Fig. 186.

The head from chimney effect may now be computed from the principles of Chapter VIII, and using a coefficient of discharge of 0.65 the

size of the openings for handling air by this method could be determined. Randall, however, has given a set of curves of flow in cubic feet per minute per square foot of opening of pivoted windows at different temperatures and heights of equivalent chimney, Fig. 187, which may be used in this work. The type of window has practically no effect, and

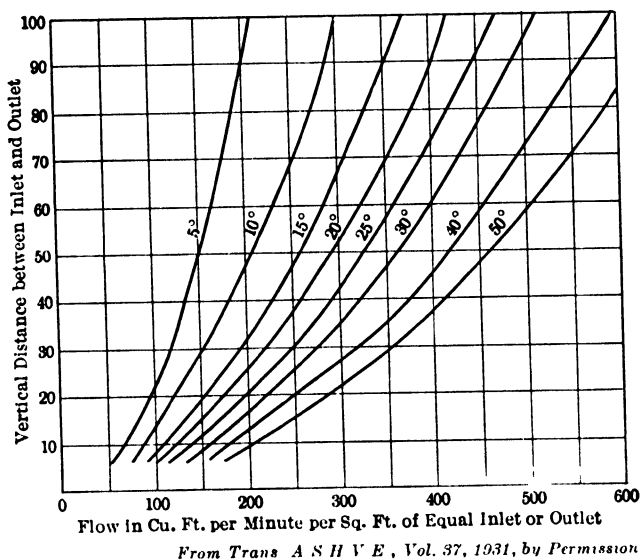


FIG. 187.—Window Ventilation from Chimney Effect..

the area would be equal to the full area multiplied by the sine of the angle of inclination of the leaf. (Temp. on curve = $t_i - t_o$.)

When both wind and chimney effect act together the combined result is given by Randall in Fig. 188 as a multiple of the chimney the effect when this factor is expressed as a percentage of the sum of two elements.

In the shop of Fig. 185 it is desired to introduce 304,000 cfm additional air. The usual wind at this site is 6 mph, and the building is placed so that the long side faces this wind. The air is so introduced and mixed with building air that the temperature difference between the air at the level of the high side window is 40° F below that at the temperature of the monitor window 14 ft above this window. From Fig. 187 the chimney effect flow per square foot of window opening would be 200 cfm. The flow from the wind perpendicular to the face with a monitor would be 60% of the wind velocity of 6 mph or 317 fpm. This would give an air flow through the windows of 317 cfm per sq ft. The chimney effect is $\frac{200}{517} \times 100\%$ or 38.5% of the sum of the effects. From Fig. 188 the combined effect is 1.75 times the chimney effect, or the inflow will be 350 cfm per sq ft. The open window area required is

14.5 sq ft or $\frac{304\ 000}{350 \times 60}$. If each of fifteen distributed windows were open to give 1 sq ft opening, and the same area were provided in the monitor, this amount of air would be admitted by these two actions. Although this area is small, all points of design have been considered except the effect of the wind should the flow be parallel to the length of the monitor. As the building is about half as long in the direction

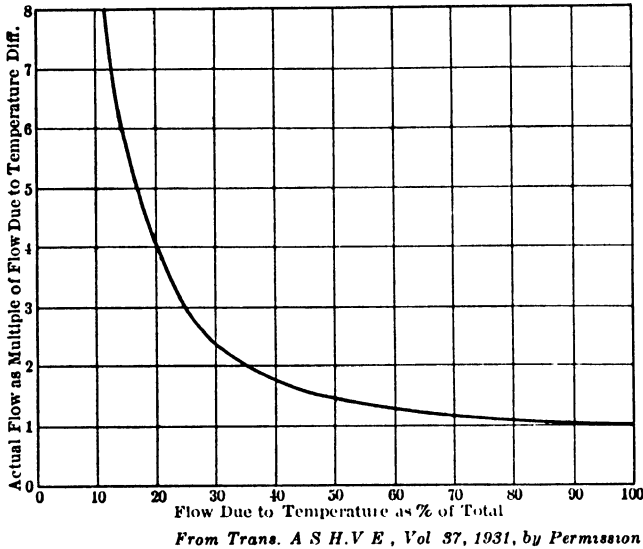


Fig. 188.—Combined Effect of Wind and Temperature.

of the wind as the width perpendicular to it, there would be a very small aspirating from the sides.

The heat demand per hour for this building, before occupancy in morning,* will be as follows:

Wall loss (including monitor).....	144 000 Btu
$15,000 \times 0.24 \times (60 - 20)$	
Glass loss.....	452 000 "
$10,000 \times 1.13 \times (60 - 20)$	
Roof loss.....	180 000 "
$18,000 \times 0.20 \times (70 - 20)$	
Air loss (approximate).....	435 000 "
$\frac{600\ 000}{55.6} (60 - 20)$	
	<u>1 211 000 Btu per hr</u>

This will be provided by heating air from 60° to 120° in two Vento heater units, one on each side of the building. Three stacks of regular sections at 5 in. centers would give 120° F with 60° supply at 1000 fpm with 1.28 lb of condensation per

* Bodily heat and that from machines and sun effect during occupancy would decrease the heat demand.

sq ft. Each heater would require 493 sq ft of surface $\left(\frac{1\,211\,000}{2 \times 1.28 \times 960.6}\right)$ or 164 sq ft per stack. The 70° F air handled in each unit with this 60° difference would be 9350 cfm $\left(\frac{1\,211\,000 \times 55.6}{2 \times 60 \times 60}\right)$. The net area between the sections of the stack at 1000 fpm will be 9.35 sq ft for 70° air.

From the table of Vento section dimensions 164 sq ft of surface and 9.35 sq ft of net air space would be obtained with ten 60 in. sections at 5 in. centers having 160 sq ft of surface and 9.21 sq ft of net air space. One stack would be 50 in. wide, $60\frac{1}{8}$ in. high and $9\frac{1}{8}$ in. deep, and would weigh 1310 lb.

The air from each unit is to be divided into four equal parts, and each of these will be divided into four parts to be discharged towards the outside wall at the center of each bay shown. Each branch will deliver 583 cfm $\left(\frac{9350}{16}\right)$ of 70° air or 638 cfm at 120° F. In this shop the velocity at the outlet will be taken as 600 fpm requiring 153 sq in. of area.

The velocities in the round ducts will be taken as 1400 fpm and in the vertical risers as 1600 fpm. These will take the following sizes from Fig. 148.

Section	Cfm at 120° F	Diameter, inches	70° Air Friction per 100'	Length	Friction
1	638	9 5	0 35	21	0 074 in.
2	1 276	13 0	0.25	21	0 053
3	1 914	16 0	0 20	21	0 042
4	2 552	18.0	0.18	10	0.018
Risers	5 104	24 0	0 16	20	0 032
Risers	10 208	35.0	0 11	10	0.011
					0.230 in.

$$\text{Actual friction} = \frac{\text{Friction at 70° F per 100 ft}}{100} \times \text{Length} \times \text{Relative density}$$

Loss in branch:	0.159	$\times 0.914 \times 0.15$	0 022 in.
Loss in elbow ($R/W = 1$):	0.122	$\times 0.914 \times 0.26$	0.029
Loss in reducers:	0.122	$\times 0.914 \times 0.04 \times 3$	0.013
Loss in branch:	0.122	$\times 0.914 \times 0.15$	0.017
Loss in outlet:	0.0224	$\times 0.92 \times (1 + 0.75)$	0 036
Loss at entrance to heater at 1000	$\times 9/23$	or 400 fpm 0.5×0.010	0.005
Loss in Vento heater			0.160

0 282 in.

Total pressure with 120° air $(0.282 + 0.230 \times 0.914)$ 0 500 in.

Total pressure with 70° air of same weight 0.500×0.914 0 457 "

Total volume at 70° F for equal weight $10,280 \times 0.92$ 9350 cfm.

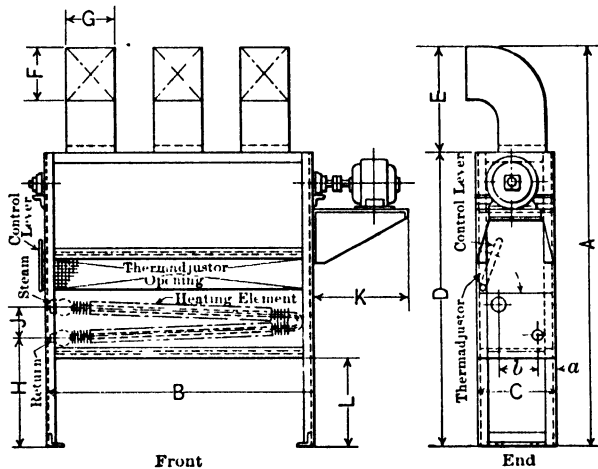
The fan in this case will draw in from the heater and discharge at 1600 fpm at 120° F and at 1600×0.914 or 1460 fpm at 70° F. The velocity head is 0.133 in. leaving a static head of 0.324 in.

From Table A-20, the Silent Vane fan No. 125 at 1475 fpm outlet by interpolation would deliver 28,400 cfm. For a static pressure of 0.324 in. the speed would be 287 rpm and the horsepower 3.00. Other tables will have to be consulted for the actual selection of the fan, but the data above can be used in connection with Table A-23 in determination of size.

$$\text{Per cent reduction in diameter of wheel} = 100 \sqrt{\frac{9350}{28100}} = 58\%$$

Diameter of No. 125 wheel $62\frac{1}{2}$ in.

Diameter of required wheel $62\frac{1}{2} \times 0.58 = 36$ in.



Courtesy of B. F. Sturtevant Co.

FIG. 189.—Rexvane Unit Heater.

This is the diameter of wheel for size No. 90.

By fan laws

$$\text{Actual speed } 287 \times \frac{62.5}{38.0} = 472 \text{ rpm}$$

$$\text{Horsepower } 3.00 \times \left(\frac{38.0}{62.5}\right)^3 = 1.11 \text{ hp}$$

These data must now be changed to those for 120° F air but of equal weight

$$\text{Speed} = \frac{472}{0.914} = 517 \text{ rpm}$$

$$\text{hp} = \frac{1.10}{(0.914)^3} = 1.32 \text{ hp}$$

If six unit Rexvane speed heaters of the Sturtevant Company as shown in Fig. 189 were used on each floor for this problem the air handled by each unit would be 1550 cfm and the heat added by each would be 100,000 Btu per hr. Reference to the data sheets of the Sturtevant Company show that size 100 with steam at 2 lb gage pressure would deliver 100,000 Btu, giving a final temperature of 121° F to 1495 cfm of 70° air at 1150 rpm of fan. It would require 0.169 hp to drive. These units could be selected and placed as shown dotted on Fig. 185 at S. S. The dimensions of these units are comparable with the units for which data are given in Table A-25.

PROBLEMS

1. Design a unit heater to care for a room needing 70 Mbh for heat loss through walls with leakage, with 50 persons at medium work, using 15% outside air at 25° F, R.H. 70%. Circulation, 2000 cfm per person.

2. This same room has 800 sq ft of wall space on west side, 6'00 sq ft of glass with awnings. The wall is 12 in. brick with furred plaster. There is no heat leakage at floor or ceiling. No lighting or power is used. The room is held at 80° F dry bulb and 70° F wet bulb when the outside temperature is 90° F dry and 85° F wet. How much fresh air would be used? How much air would be circulated? How much cooling surface would be installed? Water is available at 55° F.

3. At 70° F, 150,000 cfh of air is delivered into a room at wet bulb of 68° F. The air entered the cooler at 83° F dry bulb and 75° wet bulb. Water is available at 55° F. How much water-cooled surface would be used?

4. How much heat would be required for the shop of this chapter after occupancy?

5. How many tons of refrigeration will be required to hold the shop of this chapter at 80° F-70° F when the air outside is 90° F-85° F? Wall and glass divided in proportion to length of side. Wind from west.

CHAPTER XI

WARM AIR FURNACE HEATING

As shown in Fig. 1, buildings may be heated by air which has been warmed by the combustion of a fuel in a furnace. In furnace heating, air is discharged into leader pipes from the heating space of the furnace, and these deliver the heated air into register floor boxes just above the basement or through boots into the risers or stacks for delivery to register faces at higher levels. If this air is heated to temperatures of 175° F or less the system is called a warm air system by some; if above this temperature it is called a hot air system. As shown in Chapter I, the air supply may be taken from the outside, recirculated from the building or a mixture of outside air and recirculated air may be employed. The first and third methods are adopted for school buildings, auditoriums or churches but for residences or office buildings in which there are few persons in the occupied spaces, the air leakage from the windows will give sufficient fresh air supply so that recirculation alone furnishes a proper mixture. The warm air is circulated by natural draft or by fans.

In the design of these systems the heat requirements for each room are found as in Chapter V and the amount of air per hour to supply this heat in cooling from the entering temperature to room temperature can be found, as soon as the entering temperature is fixed by the designer,

$$q = 0.018 \times 60V(t_i - t_r) = \frac{60V}{55.6} (t_i - t_r) \quad (11-1)$$

In this formula q includes the heat to warm the leakage air as well as the loss from six faces of the room. In fact, the procedure is quite similar to the method of indirect steam heating, natural draft being similar to that of the gravity method and fan circulation to that of the plenum system.

Gravity System. With the temperatures of the warm air and that of the building known, the gravity head per foot from furnace to register face causing air flow would be known in inches of water as $7.65 \left(\frac{1}{T_R} - \frac{1}{T_i} \right)$,

and the size of leaders, risers and registers to carry the equivalent of V cubic feet per hour of standard air may be found by successive approximations, using the friction loss in pipes and fittings as given in Chapter VIII.

The extensive studies made by President Willard and his associates at the Engineering Experiment Station of the University of Illinois in cooperation with the National Warm Air Heating and Ventilating Association and reported in bulletins* have so determined the elements of performance of such systems that their findings have made the selection of temperatures, sizes and equivalents more simple and certain than is possible by theory alone. These results have been freely used in the following discussion.

It is quite evident from Chapter VIII that the heads causing flow produced by a few feet of warm air are very slight and for this reason

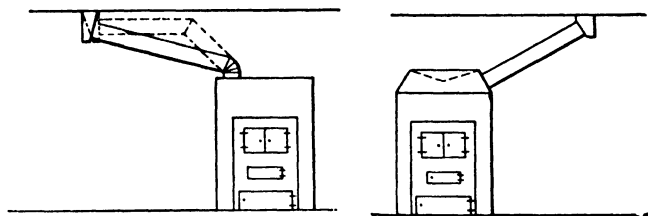


FIG. 190.—Warm Air Furnaces.

this natural draft system is not applicable to buildings with leader lengths much greater than 12 or 15 feet; when longer leaders are required fan circulation is used.

The leaders are taken from the sides of the conical bonnet at the top of the furnace casing, Fig. 190, when head room does not permit one to use a flat bonnet top which gives better circulation, provided the top is located at least 10 in. above the top of the furnace radiator. If the top of the bonnet is within 12 in. or less of an inflammable floor the Code of the National Warm Air Heating and Ventilating Association requires a metal shield held 2 in. from the woodwork and extending 18 in. beyond the casing. In no instance may the top of the casing be closer than 6 in. to the wooden joist.

The leaders are connected to collars 5 in. long extending from the side or the top of the bonnet. The tops of all these collars, regardless of size, are placed on the same level to offer equal resistances to the flow, and the leaders are pitched upward at least 1 in. per ft. This is of

* See Bulletins 112, 117, 120, 141, 188, 189, 230, 246, 266 of the Engineering Experiment Station of the University of Illinois.

value when circulation is initiated, and it also prevents currents in two directions which may occur in horizontal pipes of these systems with small driving head. All leaders except one should be furnished with a damper with two supports, placed within 2 ft of the bonnet, Fig. 191, for the purpose of regulating the warm air distribution among the various circuits. It is necessary to have at least one circuit which cannot be closed on account of the fire hazard resulting from the high temperature which might be found within the casing were no air circulated. With

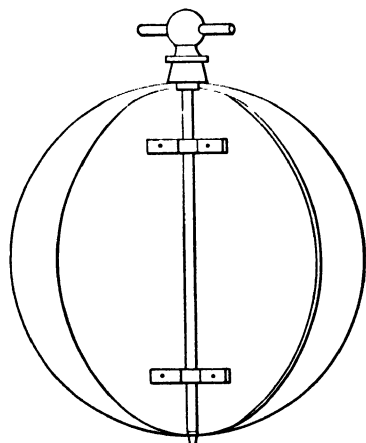


FIG. 191.—Damper.

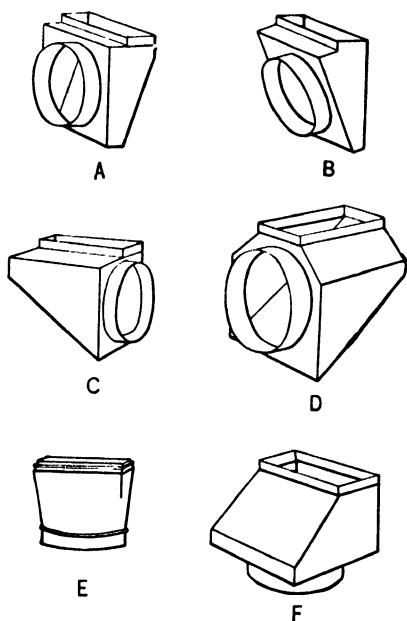


FIG. 192.—Boots.

one outlet only as in a furnace for a church or in a so-called pipeless furnace, no damper is used.

The leaders are usually made of bright tin or galvanized iron, with asbestos paper around the slip joints between sections. It has been shown that paint or asbestos paper placed around the complete leader pipe increases the heat loss on account of the greater loss from radiation when the bright tin is covered. This is true even when as many as six layers of 12 lb asbestos paper (12 lb per 100 sq ft) are used. Three $\frac{1}{4}$ in. layers of asbestos air-cell insulation will reduce the heat loss by about 55%.

In the design of the duct system, leaders should never be less than 8 in. in diameter and elbows should be reduced to a minimum. The

boots, Fig. 192, connecting leaders to the stacks, are formed so as to reduce the friction at such points and to save extra fittings. *A* or *B* would be used when the leader is at right angles to the partition; *C* or *D*, when the leader is parallel to it. *E* and *F* are employed in connection with elbows. The boot *C* is necessary to offset the leader to one side of the stack.

When the leader is carried to the floor register just above the basement, the floor box, Fig. 193, with no projecting fins is used. These boxes are from $4\frac{1}{2}$ to 6 in. deep to allow for circulation to all parts of the outlet. The figure illustrates an elbow substituted for a boot.

Stacks are placed within the inside partitions of a building because of the loss of heat if placed in outer walls which would reduce the temperature and cause a diminution of flow and also because down drafts might occur in such lines from the cold air on the outer walls of a room.

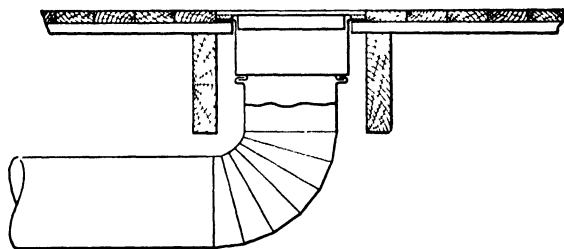


FIG. 193.—Floor Register Box.

Friction must be reduced to a minimum in the gravity system. In Fig. 190, the dotted leaders, although cheaper to install because of the use of standard fittings, would not be desirable because of extra friction. Even though extra cutting is necessary to fit parts for the reduction of friction, this must be done regardless of expense. The success of the system depends on this reduction.

When one line may have more friction loss than the others its leader is sometimes carried into a hood within the bonnet which will direct a portion of the heated air into this circuit. Leaders to the exposed parts of the house should be made shorter than others if permissible, although the attempt is made to place the furnace so that all leaders are as short as possible. The diameters of the longer leaders are made larger than the sizes required by calculation.

The available space for stacks between the studs and plaster of a partition is 3 or 4 in. in depth by 14 in. in width. To reduce the fire hazard from stacks the wooden studs, from which the stack must be separated by at least $\frac{5}{16}$ in., must be faced with thin sheet metal, the

stacks covered with 12 lb asbestos paper and metal lath used over this stack space. The stack joints must be lock-seamed and soldered, and they are held in proper position by straps. When stack walls are made of two thickness of metal with a continuous $\frac{5}{8}$ in. air space between and combined with fittings of double walls it is not necessary to cover the studs with metal and the stack may fill the available space. The space around stacks or fittings at the first floor must be filled with asbestos cement or other non-combustible material to prevent any spread of fire from the basement. These limitations on stacks are such that stack areas of only 40 or 45 sq in. are available with ordinary partitions. This means that a 6 in. partition must be used at times,

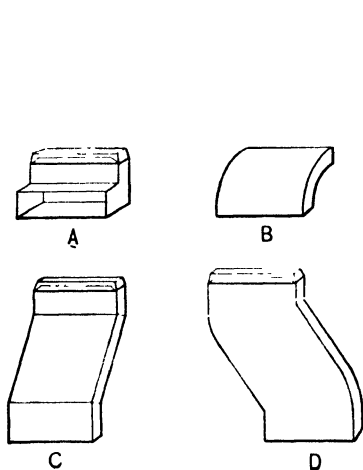


FIG. 194.—Elbows and Offsets.

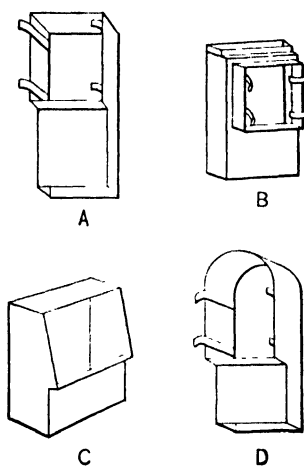


FIG. 195.—Register Boxes.

and even special flues of greater space have been built in projection parts of partitions.

Where obstructions occur in the desired stack space, offsets or elbows, Fig. 194 are used, *A* and *B* with a floor box in an upper story, and *C* and *D* when a girder or change in stud centers prevents a straight alignment.

The register boxes, Fig. 195, are used at the ends of stacks, and it is advisable to have an inclined partition or back in these to direct the air outward with little friction. The damper behind the grille or register face may be a flat plate hinged at the top for this purpose acting as a splitter to direct the air into the room. To accommodate a larger stack area and to reduce friction a special baseboard register, Fig. 196, is used. This fits over the register box *C*, Fig. 195. It projects several inches beyond the baseboard. This is a simple installation for a room on the first floor.

A general arrangement of bonnet, leaders, shoes, stacks, register boxes, floor box and special baseboard register is shown in Fig. 197.

It has been found that baseboard registers are more effective in reducing the variation of temperature between floor and ceiling than wall registers placed several feet above the floor. This variation of temperature with baseboard registers will be 14° in zero weather and $5\frac{1}{2}^{\circ}$ F in milder weather (45°) with the temperature at the breathing line about two-thirds of this difference above that of the floor. With side wall registers the temperature variations are 22° and 10.5° F, the ceiling temperature being $15\frac{1}{2}^{\circ}$ and $7\frac{1}{2}^{\circ}$ above that of the breathing line. The increased flow of heated air from these high wall registers accounts for the high ceiling temperature and greater heat losses from the walls. Baseboard registers for upper floors are found to be more effective at these points than floor registers in increasing heat flow and temperature

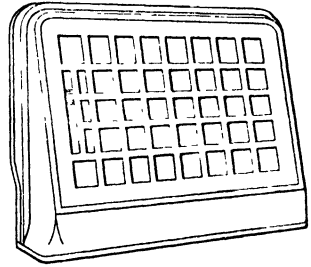


FIG. 196.—Special Baseboard Register.

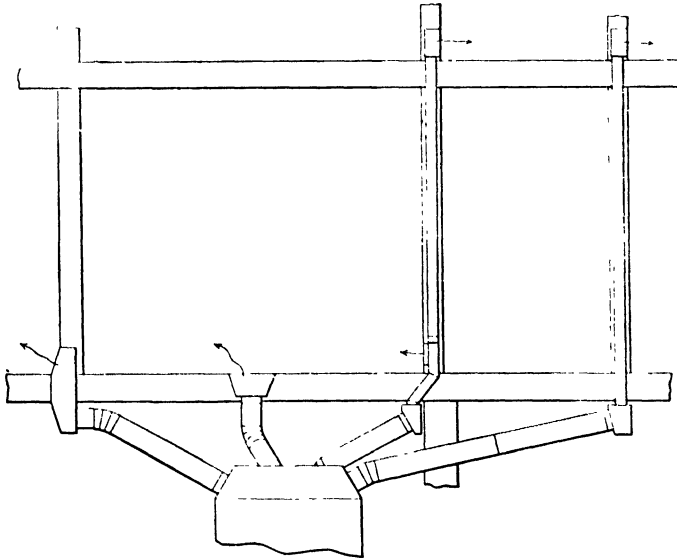


FIG. 197.—General Arrangement of Leaders, Boots, Stacks and Boxes.

because of decreased resistance, although for the first story the floor register is more effective.

When the door of a room is closed the temperature difference between

floor, breathing line and ceiling becomes less but there is a slight increase of room pressure which impedes the flow of warm air and the heat transmission.* With fan circulation louvres in baseboard register to deflect air 15° downward reduce the customary temperature difference between floor and ceiling.

The thickness of tin used for leaders and stacks is designated by the weight of 100 sheets, 14 in. × 20 in., or by letters. Thus, a 56 lb tin sheet will weigh 56 lb per 100 sheets of the size given. IC tin weighs 107 lb per 100; IXL, 128 lb; IX, 135 lb; IXX, 155 lb, up to IXXXXX, 215 lb. IC, IX and IXX tin are employed for warm air pipe work. At times galvanized sheet iron may serve for leader pipes. Nothing less than IC tin or similar galvanized iron is suitable for small sizes, and 12 in. leaders require at least IX tin or #26 galvanized iron. (See Table A-23.)

Design of System. By reference to the general data sheet, Fig. 96, it is seen that the building of Chapter V requires 115.0 Mbh for the heat losses including the heat to care for air leakage. If t_1 be taken at 175° F for a building at 70° F the air needed would be

$$\text{cfm} = \frac{115,000 \times 55.6}{60 \times (175 - 70)} = 1010 \text{ cfm}$$

If all this air were taken from the outside it would have to be heated from zero degrees, requiring an additional amount

$$\frac{60 \times 1010}{55.6} \times (70 - 0) = 76.6 \text{ Mbh}$$

This would increase by 66% the amount of heat required to supply the heat loss.

The heat loss from the building includes the warming of at least 100 cfm of leakage air, an amount of fresh air required for 3 to 7 persons.

Should the air inlet temperature be reduced to 122.5° F, twice as much air would be circulated, and if this were outside air the heat to bring it to 70° in zero weather would be 153 Mbh, or 133% of the heat loss from the building. Thus it is seen that for residences or offices full outside air means a great expense which is not necessary, and for such installations it is well to take all the air to be heated from the building itself.

The experiments at the University of Illinois have shown that, although recirculation return through grilles in a large number of rooms in a residence gave a slightly better distribution of air between floor and

* Bulletins 246 and 266, Engineering Experiment Station, University of Illinois.

ceiling, so much of the air from the upper rooms was returned through the spaces beneath and around doors into the halls and down the stair wells to the return grilles on the first floor that it is advisable to take all recirculated air from return grilles on the first floor. This air is conducted to the base of the furnace in the most direct way with as little friction as possible. In systems designed by the author, this return grille is placed in the gib panel of the stairs, Fig. 198, or in the floor of the hall beside the stairs. This latter position is objectionable for sanitary and personal reasons. If a second return grille is used it would be well to place it at the cold side of a first-floor room. The area of a

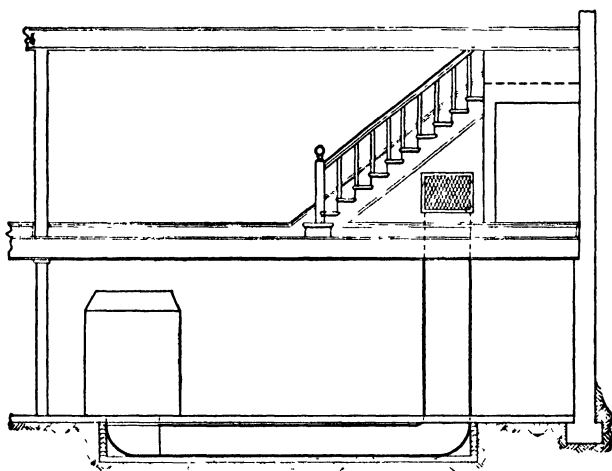


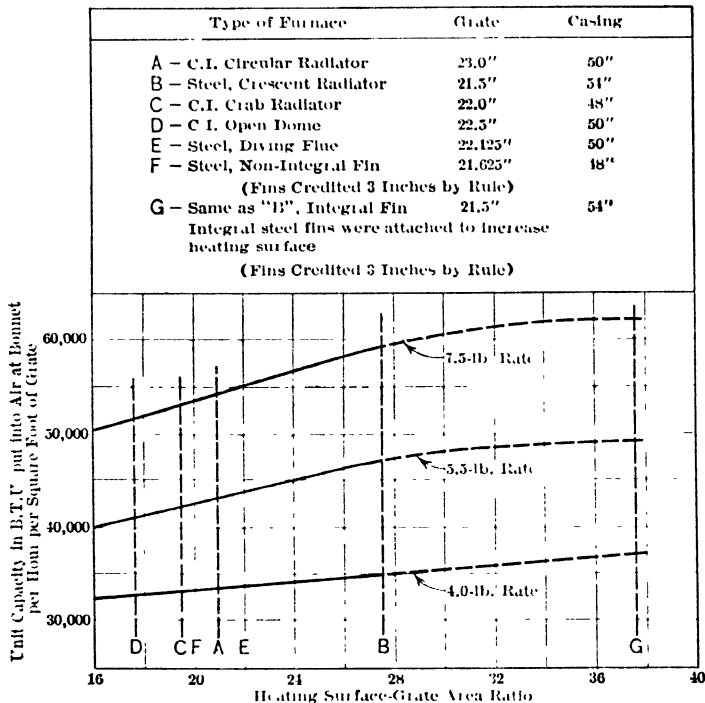
FIG. 198.—Return Grille in Jib Panel.

single return duct or pipe and the net area of its grille should be equal to the sum of the areas of the leader pipes, and where two or more returns are used their combined area should be slightly greater than that of a single duct.

Whenever resistance due to the return or fresh air supply ducts is added to a system the increase of friction means a decrease of air flow at a given temperature and a reduced heat discharge. Hotter air must be delivered to obtain more driving head, and this results in a lower furnace efficiency. For this reason all connections should be made with easy bends.

The early tests at the Experiment Station were made on warm air furnaces having ratios of heating surfaces to grate areas of as much as 28, while a later bulletin included one furnace in which the ratio of 37.5 sq ft of heating surface per square foot of grate area was obtained by

adding fins extending into the air stream. Fig. 199 shows the relation of heat put into the air in Btu per square foot of grate and the ratio of heating surface to grate area at three rates of combustion. Since the coal had a heating value of approximately 12,800 Btu per lb, the efficiency of the furnace *B* at the 4 lb combustion rate was 67%.



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FIG. 199.—Variation of Capacity of Warm Air Furnace with Heating Surface per Square Foot Grate.

The results of the tests on the system in an experimental house with 20.9 sq ft of heating surface per square foot of grate is reported in Fig. 200. This shows the variation of efficiency, hourly heat added to air in bonnet, temperature at registers with return air entering at 65° F and draft with rate of combustion per square foot per hour. The actual amount of coal burned per hour on the grate area of 2.88 sq ft may be computed. The efficiency represents the part of the heat of the fuel which is put into the air, but much of the heat supplied to this house came from the exhaust gases in the smoke pipe and chimney. These

were placed in the center of the house. This fact leads the designer of such systems to reduce the heat demand for his rooms in designing leaders and stack unless the flue and chimney are efficiently insulated.

The losses of temperature in bright tin leader pipes of different length for three different temperatures of air at first- and second-floor registers

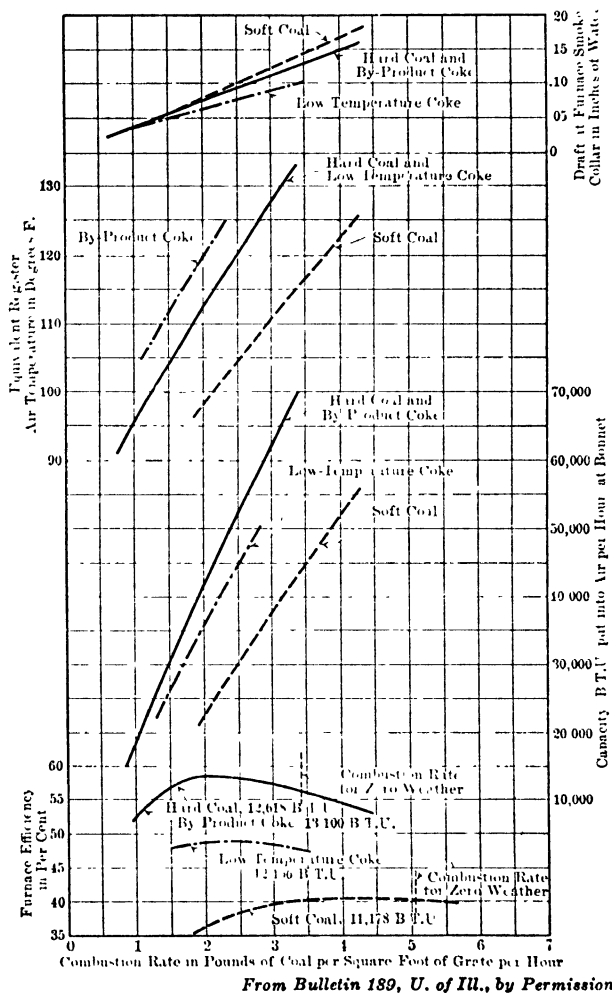
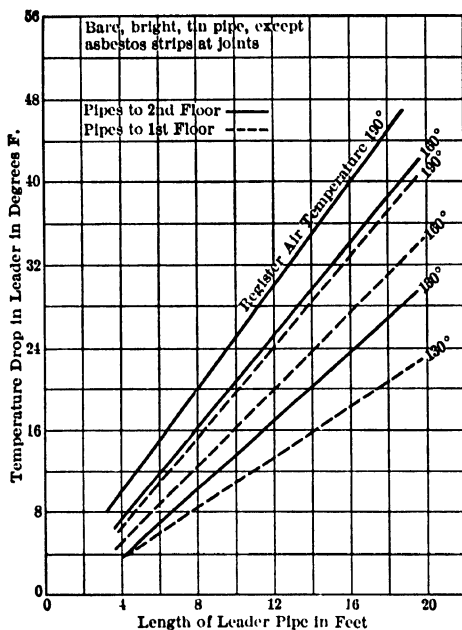


FIG. 200.—Warm Air Furnace Performance in Building.

resulting from a laboratory test are shown in Fig. 201; those in the stacks to the second floor, in Fig. 202, and the loss in stacks to the third floor is given in Fig. 203.

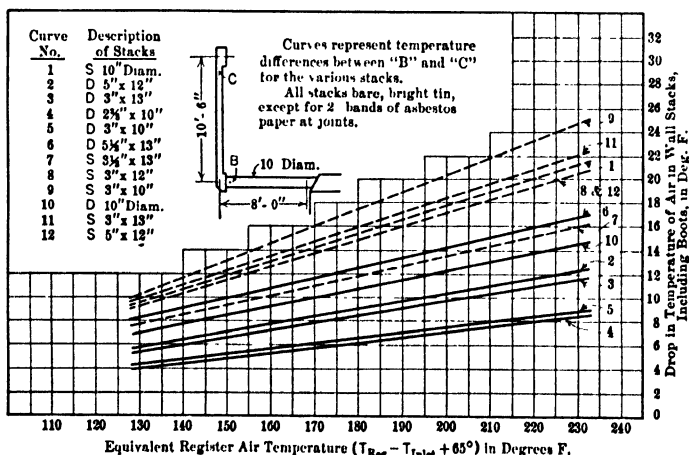
On account of the decrease in temperature the heat delivered to

various floors for the same bonnet temperature will be different since the additional flow due to the greater driving force from the stack of



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FIG. 201.—Temperature Drop in Leaders.

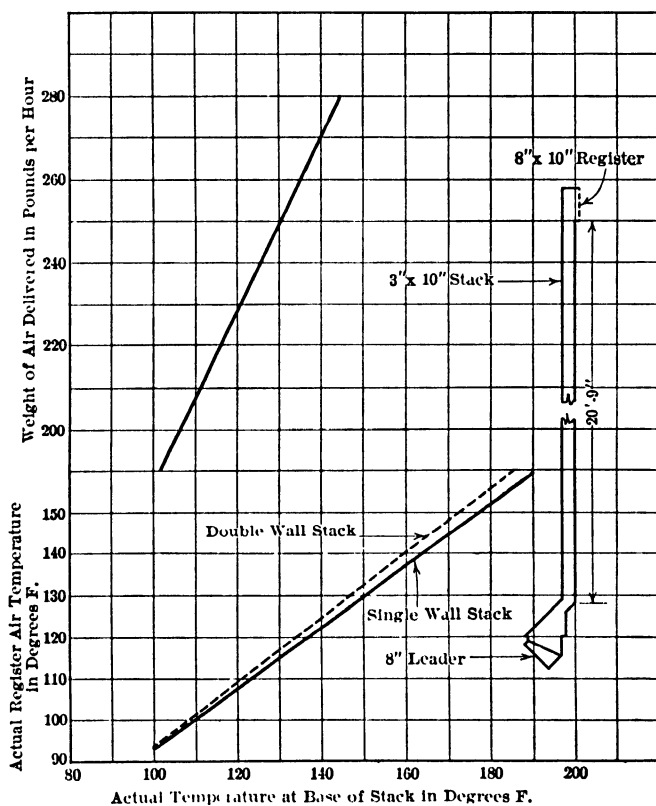


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FIG. 202.—Temperature Drop in Stacks.

greater height is partially offset by the greater drop in temperature of the air.

The experiments at the University of Illinois, however, have been reported in the amount of heat delivered into the room per square inch of leader pipe with stack areas equal to 70% of the leader area for different floors and with different register temperatures as shown in Fig. 204. Points on each line which come from the same rate of combustion are joined by dotted lines. These are points of the same bonnet tempera-



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FIG. 203.—Performance of Third Floor Stacks.

ture and indicate the way in which the temperatures of the registers on different floors will vary. When, for instance, the temperature at a first-floor register is 140° F, that on the second floor will be 131° F and that on the third floor 129° F. The heat available at registers per square inch of leader area for these three floors is 51 Btu per hr, 72 Btu per hr and 80 Btu per hr, the greater circulating heads for the upper floors drawing such greater quantities of air per square inch of leader pipe

that, even with the reduction of temperature, the available heat is increased.

The actual delivery for stacks for the second floor of areas which are not less than 70% of the leader areas has been actually measured in the laboratory for $10\frac{1}{2}$ ft stacks connected with 10 in. and 8 in. leaders each 8 ft long. These results are given in Fig. 205, and the effect of height of

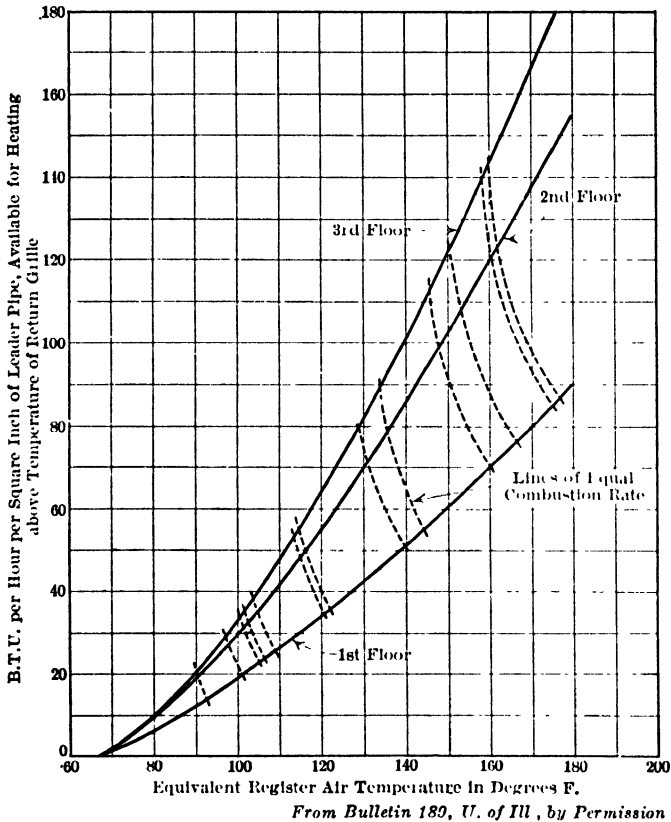


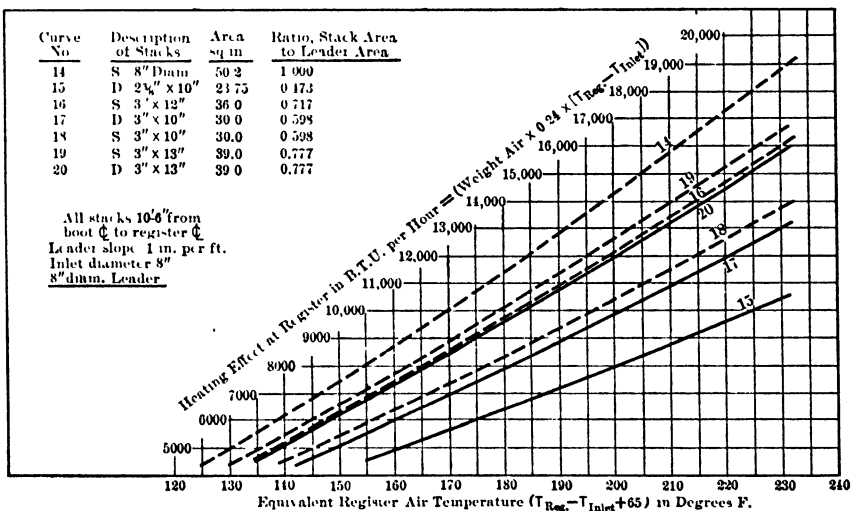
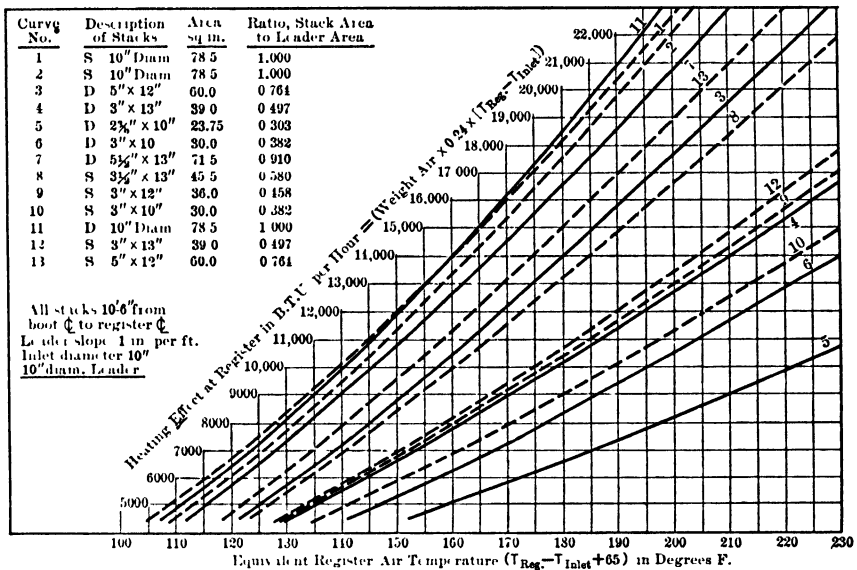
FIG. 204.—Heat Delivery and Temperature at Register.

$4\frac{3}{8}$ in. \times 13 in (inside) stacks of double walls is reported in Fig. 206, for four different register temperatures. This figure gives one curve of output for a given heater input which indicates that stacks of 10 to 12 ft height are the most effective.

For first-floor baseboard registers the stacks must have an area equal to the leader area.

Tests on a number of different furnaces indicate that the free area for air passage within the casing should be 46% of the total horizontal cross-

sectional area of the casing and the sum of the leader areas should be 74% of the free area or 34% of the gross cross-sectional area of the cas-



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FIG. 205.—Heat Delivery from Second-story Stack on Two Sizes of Leaders.

ing. Good results were even obtained with a free area of 37% of the gross area and a leader area of 110% of the net area, although, with leader areas greater than 100% of the free area, there is a possibility that

certain leaders may become supply ducts. The velocity occurring with the smaller percentage prevents this reversal, which is called breathing by Willard and his associates.

These tests have indicated also that increased efficiency and capacity

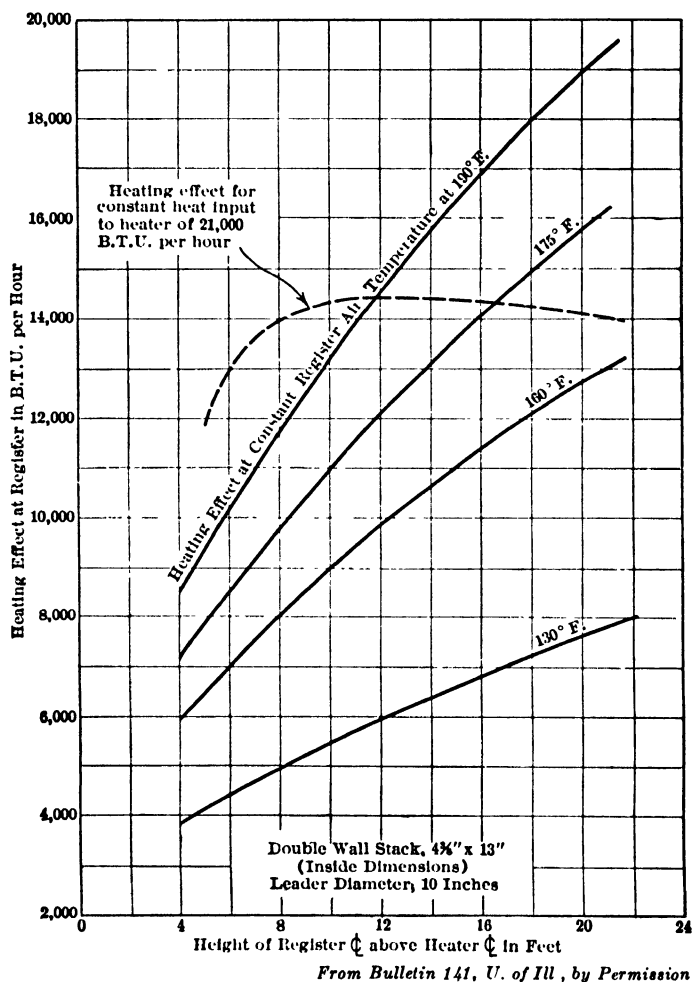


FIG. 206.—Effects of Stack Height.

of furnace result from the use of 2 in. insulation applied to the bonnet and cast-iron front of a casing equipped with a lining within the casing and from insulation on the floor surrounding the ash pit. The saving amounted to 11.8% when register temperatures were 130° F and to 9.3% when temperatures were 190° F.

A lining of black sheet iron held at 1 in. from the galvanized casing reduces the heat loss through the casing from 6% to 2.75%, and when an extra shield is introduced in the free air space the casing loss is reduced to 1%. On account of the small casing loss it is unnecessary to cover the casing.

These casings are made of galvanized iron of No. 26 U.S. gage or heavier. All parts must fit the cast fronts and casing ring in order to be dust tight. See Table A-23 for the thickness of sheets.

The warm air furnace may be mounted above a pit formed of masonry for the admission of air from the return duct, or on a solid foundation with air inlets arranged in the sides of the casing.

Design of System. The data from the Experiment Station will be used in the design of the warm air furnace system for the building of Chapter V.

From Fig. 200 it is seen that in an actual building system the efficiency of the simple cast-iron radiator furnace with 21 sq ft of heating surface per sq ft of grate will have its greatest value of 58% with hard coal when the rate of combustion is 2 lb per sq ft of grate per hr, whereas with bituminous coal high efficiency of about 40% extends from $3\frac{1}{2}$ lb to $5\frac{1}{2}$ lb of coal per sq ft of grate per hr. The register temperatures will be taken as 112° F for hard coal, and they extend from 108° to 138° F with the soft coal. At a combustion rate of 5 lb per sq ft of grate per hr, with hard coal, the efficiency drops 52% and the register temperature becomes 160° F by extrapolation. The draft needs for these various rates of combustion are 0.07 in. and 0.18 in. for the hard coal and 0.13 in. and 0.22 in. for the bituminous coal.

The low air heating efficiency for these furnaces is due to the small heating surface, and were more surface used than that of furnace A (22.9 sq ft per sq ft of grate), the efficiency would have been higher as shown by Fig. 199. The house efficiency in this plant is above 90%, as much heat was delivered from the leaders, stacks, smoke pipe and chimney.

It is now decided that, in zero weather with hard coal, the rate of combustion will be 5 lb per sq ft per hr as the efficiency is not reduced too much and the 40 ft chimney height will give the necessary draft with the hot flue gases at such periods.

For the average register temperature of 160° F, from Fig. 204, the first-floor register will be 170° F and the second-floor register will be 154° F. The heat available per hour per square inch of leader pipes from the same figure will be 80 Btu for the first floor and 115 Btu for the second.

Using the heat losses given on the general data sheet, Fig. 96, the following table results for leaders, stacks and registers, using the stack areas of 100% of leader areas for the first floor and 70% for upper floors, while the net register areas are equal to 100% of the calculated leader area. No leaders are made less than 8 in. in diameter. The free area of the register is taken as $66\frac{2}{3}\%$ of the gross area.

Since the loss from the leaders and stacks amounts to about 30% of the heat in the bonnet and this is used for heating the house, only two-thirds of the heat loss in each room will be cared for by available heat from the leader.

The Code for Installation of Gravity Warm Air Heating Furnaces uses empirical rules equivalent to 110 Btu per hr per sq in. of leader area on the first floor with no allowance for heat radiated from leaders and stack, 160 Btu per hr per sq in. for leaders to second floor and 190 for third-floor leaders.

TABLE 11-1
WARM AIR SYSTEM FOR RESIDENCE OF L. Q. SMITH

Room	Mbh	Leader Area, sq in.	Leader Required	Diameter Used	Stacks Required		Registers Required	
					Area	Size Used	Area	Size Used
101	19.7	165	14 5	15	165	Floorbox	267	15×16
102	14.3	119	12 4	13	119	"	200	12×16
103	2 1	18	4 7	0	0	0	0	0
104	6.5	54	8 3	8	54	Floorbox	75	8×10
105	10.0	83	10 3	10	83	"	120	10×14
106	12 4	102	11 4	12	102	"	160	10×16
201	15.2	90	11 0	11	63	5×13	143	12×12
202	11.0	64	9 1	9	49	5×12	97	10×10
203	3.5	21	5 2	8	15	3×6	75	9×9
204	8 7	50	8 0	9	35	3×12	97	10×10
205	8 8	50	8 0	9	35	3×12	97	10×10
206	2 8	16	4 5	8	12	3×6	75	9×9
	115 0	832						

These leader and furnace arrangements in the basement of this building are shown in Fig. 207, and the room registers are indicated in Fig. 208. The sum of the actual areas of the leaders is 939 sq in. and that required is 832 sq in., so that 30 in. × 28 in. return duct is sufficiently large although the strict interpretation of the code would require a duct 30 in. × 32 in.

The smoke flue and furnace will be discussed in Chapter XII.

In computing the losses from the rooms the designer may pay attention to the fact that in zero weather the floor temperature drops to 60° and in 30° weather to 65° with a 70° breathing-line temperature and use a higher breathing-line temperature for this reason. For zero weather the breathing-line temperature might be taken as 74° F, the ceiling as 78° and the floors at 64° F. In milder weather these would be 70° F, 72° F and 67° F.

Warm Air Furnace Heating with Forced Circulation. To force air through longer leader pipes and also to make circulation more positive, fans have been added to the system. The investigations at the University of Illinois have included the use of centrifugal fans and propeller fans placed in the inlet duct system as well as propellers in the bonnet of the casing. These tests indicate that, with greater circulation of air, the efficiency and the capacity of the furnace have been increased and the register temperatures have been decreased. The balance of distribution for natural draft circulation is disturbed, however, when the fan

is started, as the short first-floor circuits of large area with little natural driving force now receive an excessive amount of air and the circuits to the upper floors which were made small because of the greater driving

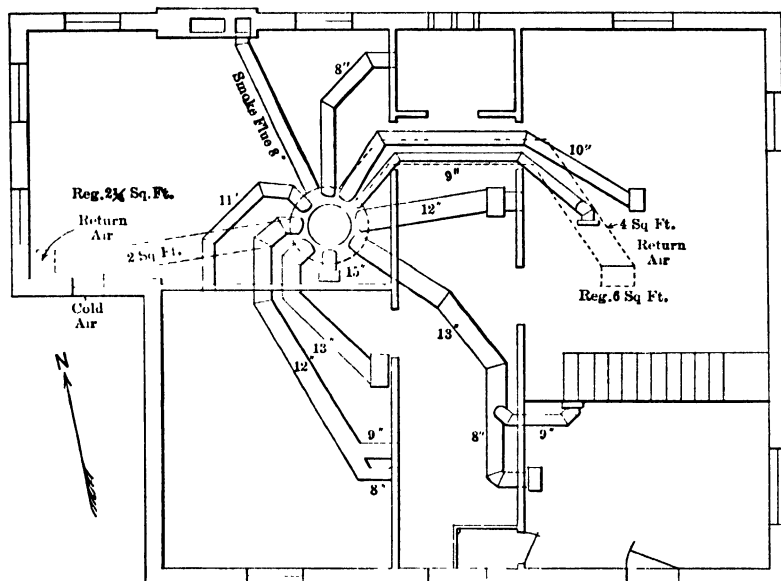


FIG. 207.—Cellar Plan for Warm Air System.

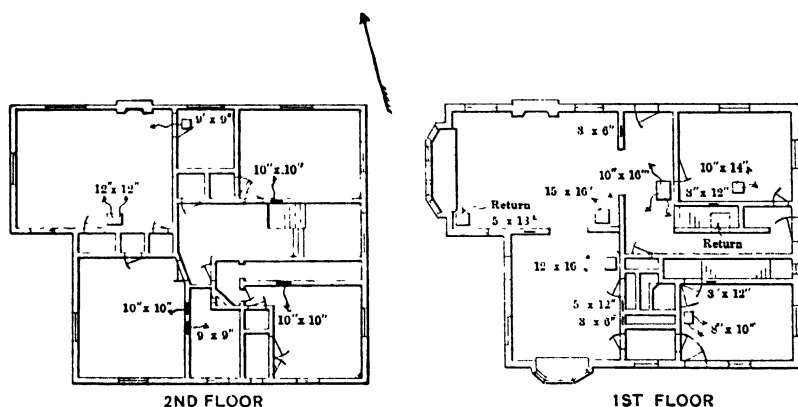


FIG. 208.—Warm Air Register Layout.

heads have a deficiency of air and heat because of their length with its accompanying friction. If the system is balanced by design or by dampers for the fan circulation, it is unbalanced as soon as the fan is shut down by hand or by an automatic control. When this happens the

first-floor registers receive less air than the registers of the upper floor. Moreover, with the fan shut down, the fan offers extra resistance and interferes with proper circulation. For the heating of small buildings, the increased furnace efficiency reduces the heat given to the building from the pipes and chimney so that the over-all efficiency of the system is not greatly changed and the small cost of the electrical energy needed to drive the fan may be greater than any saving in the furnace itself. For this reason there is little to favor the use of the fan system in buildings which have no long leaders and no extra resistances requiring fan pressures.

When, however, warm air furnaces are to be used in schools, churches or other large buildings, a fan system such as shown in Fig. 209 is neces-

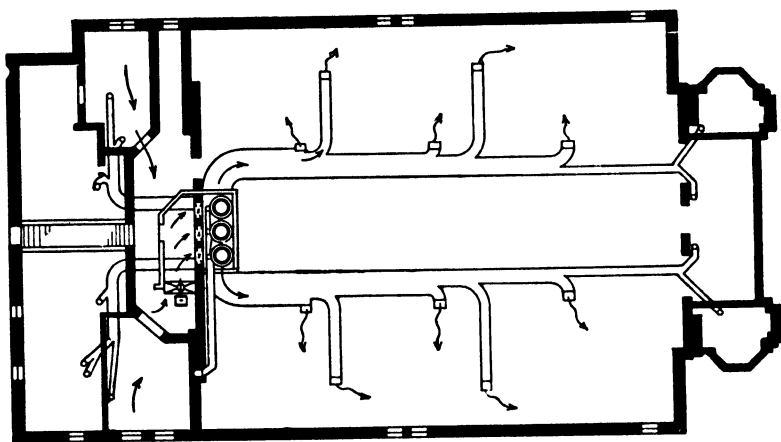


FIG. 209.—Warm Air Furnace with Fan Circulation.

sary. A number of modern warm air heating furnaces have been designed for systems in which fan circulation replaces gravity circulation on account of the addition of conditioning devices. In these systems, the arrangement of ducts, their sizes and the necessary fan sizes are determined by the methods described in Chapter IX. The velocities are now under the control of the designer and ducts may be made of sizes to fit available space, the limitations of the small head gravity system being removed. Although for the fan system the Code of the National Warm Air Heating and Air Conditioning Association gives certain empirical rules for the determination of volumes of air and sizes of ducts, as well as fan characteristics based on theoretical relations, it is well to follow the methods previously described and use the empirical rules as checks.

Humidification may be added with forced circulation although for years furnaces have been equipped with water pans for this purpose. Little moisture was added by these earlier water pans because of their limited size, the frequent failure to supply water to the pans and their poor location for evaporation. The experiments of Willard and his associates have shown where such pans should be located and the amount of water needed for comfort. The desired effective temperature in the comfort zone fixes the amount of moisture per pound of air in the building, and the possible moisture content which can be carried without window frosting has been given in Chapter IX. The designer can, therefore, compute the amount of moisture per hour which must be added to the fresh air which enters by leakage and by the fresh air inlet, if used, to bring the moisture content of the building air to the desired point. The pans must be arranged to care for this evaporation, and the Illinois tests give an evaporation of 0.04 lb of water per hr per sq in of surface when the pan is placed in the bonnet. By placing a pan around the firepot, 0.03 lb of water may be evaporated per sq in. per hr. Automatic devices maintain the water level. With fan circulation other methods of humidification are available.

PROBLEMS

1. A building with warm air heating has the following heat losses in Mbh:

1st Floor		2nd Floor		3rd Floor	
Room	Mbh	Room	Mbh	Room	Mbh
101	25	201	10	301	25
102	10	202	15	302	20
103	15	203	12	303	30
104	25	204	15	304	20
105	20	205	15		

Find the sizes of leaders, stack, registers.

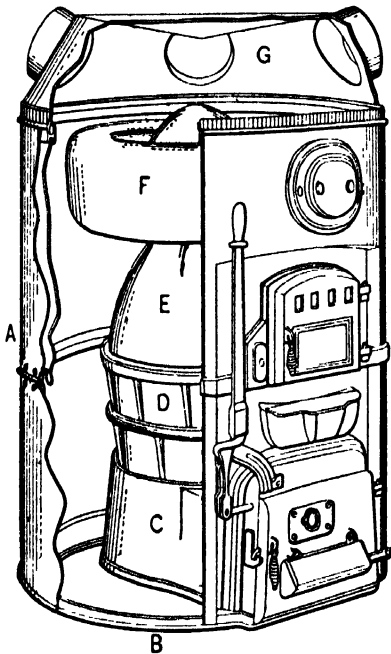
What thickness of metal should be used for leaders and stacks? Can all stacks be placed in partition spaces using 2 in. \times 4 in. studs?

2. The building above uses air at an average of 160° F with building at a temperature of 67° F to care for heat loss and 2500 cfm of leakage, with 10° F outside air at relative humidity of 50%. How much heat is saved by recirculation over that required if all air circulated is taken from the outside? How much moisture would be added per hour to maintain the humidity at 40% with the leakage air of 2500 cfm as given? Would storm sash be needed with this air to prevent frosting?

CHAPTER XII

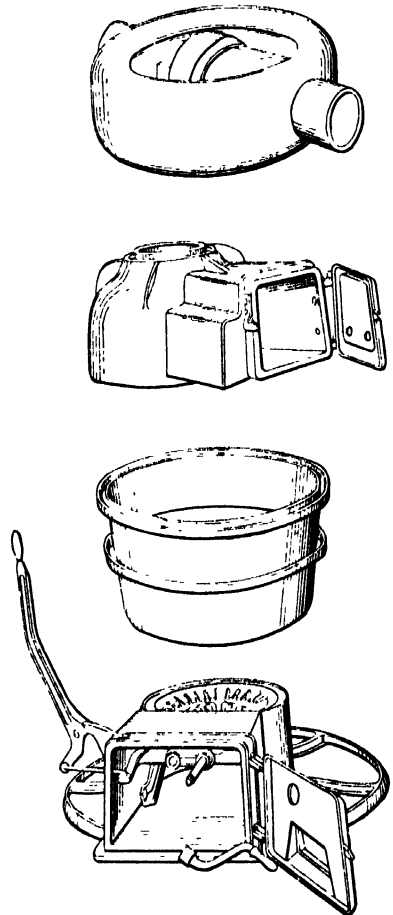
FURNACES, BOILERS, HEATERS

One form of warm air furnace, Fig. 210, is built up of four castings supported on a foundation and surrounded by a galvanized sheet-iron casing *A*, which is made of two thicknesses of metal with a circulating air space between. This inner lining extends over the radiating part of the furnace casting for heat insulation and extra heating. The base, *B*, may be placed above the air duct passing beneath the basement floor



Courtesy of L. J. Mueller Furnace Co.

FIG. 210.—Warm Air Furnace.



Courtesy of Hart and Crouse, Inc.

FIG. 211.—Cast Iron Sections for Warm Air Furnace.*

or the duct may enter the side of the casing above the floor. The casting *C* with a door forms the ash pit and supports the shaking and dumping

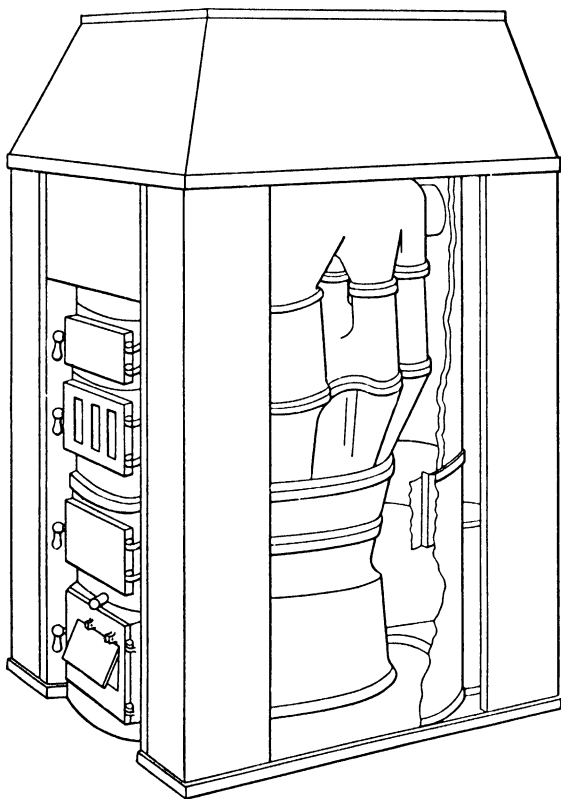
grates. The firepot castings, *D*, are made of two parts, and in some cases air inlets in the lower section supply part of the air for combustion from the air passing through the furnace. The dome or combustion chamber, *E*, contains a horizontal neck on one side leading to the firing door and another neck at the top leading to the radiator casting, *F*, above. These separate castings as made by one manufacturer are shown in Fig. 211. The bonnet, *G*, with the inverted cone top and with three branch collars caps the casing. The cover of the water tank for humidifying the warm air is shown on the cast-iron front below the feed door. The cast sections are joined by means of a tongue on one casting which fits into a groove on the adjacent section, and this is filled with stove cement. This cement is a mixture of asbestos and hardening materials which becomes set after heating and forms a gas-tight joint.

The hot gases of combustion enter the radiator from the top of the combustion chamber and flow around the complete radiator to the back. These castings transmit heat by conduction to the air passing over them in the passage from base to bonnet, and radiant energy is absorbed by the casing which then heats the air by contact. Experiments have been made by adding an extra radiation shield of metal placed in the air passage which resulted in an increase of furnace efficiency due to added air heating surface. The radiator is equipped with a capped neck on the opposite side from the smoke outlet for cleaning. The cleanout door is shown at the top of the cast-iron front above the other doors.

The grates shown are made up of circular and annular sections, but at times four straight bars mounted on trunnions at front and rear may be used. These are connected in pairs by gears so that on turning one bar the adjacent bar on one side turns in the opposite direction. In this way one-half of the grate is shaken at one time.

To increase the heating surface, the combustion chamber has been made of two sections: one for the lower part with the firing door, and one with a series of necks to be connected to a double radiator casting with a series of necks on the lower face, Fig. 212. This particular furnace has a circular casing which is surrounded by an insulated casing of flat surfaces. The return air duct enters the side of the casing above the floor. The figure illustrates the double cylindrical casing and also a hanging radiator shield. The heating surface is further increased a slight amount by casting the firebox section with vertical convolutions. Another method of construction is to make the ash pit, firebox, combustion chamber and the side radiator of sheet steel, Fig. 213. An inner partition *A* within the kidney-shaped radiator forces the gases to take a long path through this portion of the furnace. In many furnaces a damper, such as that at *B*, gives a direct path from the combustion

chamber to the flue for use when starting fires, or forcing them. The firebrick lining, the two-part grate, the double casing as well as the inverted cone bonnet are shown. The air to be heated enters the casing from a duct below the floor. The water tank is at *C*. The tank may extend over so much area that proper humidification occurs especially if the water supply is controlled by a float. Special water pans of large



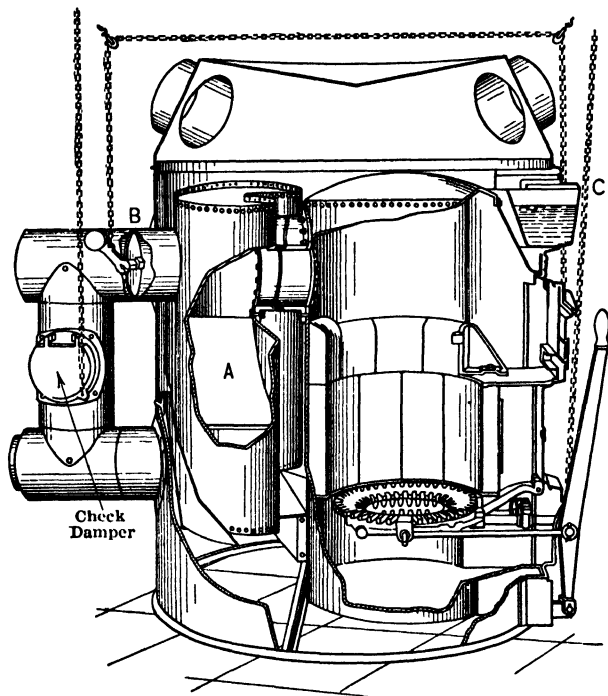
Courtesy of L. J. Mueller Furnace Co.

FIG. 212.—Double Radiator Warm Air Furnace.

size and even steam from pipe coils in the combustion chamber are used to humidify the air.

The Kelsey warm air generator, Fig. 214, is one in which a large amount of prime heating surface is exposed to the air to be heated. In this, the firepot and the combustion chamber are formed by a series of hollow cast-iron, wedge-shaped members which form the circular firepot and extend beyond as heating flues. These sections are so separated

above the fuel level that three sides of the flues are exposed to the hottest gases and to radiation from the fire. Fins on the outer surfaces so meet that the furnace gases are forced to travel to the top of the sections, at which points openings allow them to pass to a space around the fourth outer side of each section. Here the gases are conducted downward by a sheet-steel casing forming the so-called combustion space.



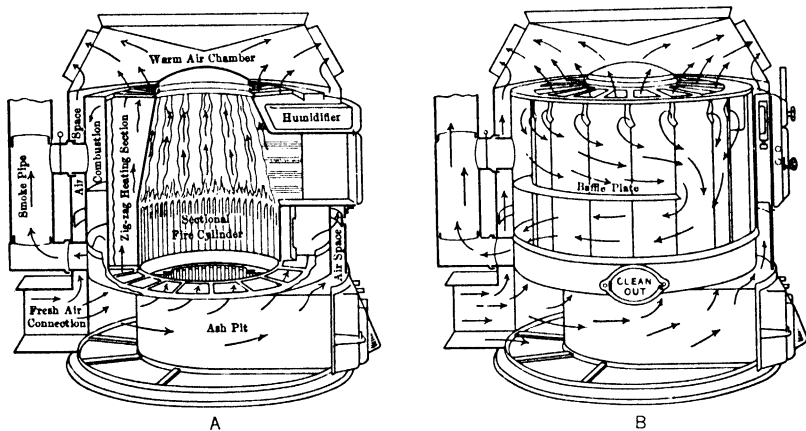
Courtesy of L. J. Mueller Co.

FIG. 213.—Steel Warm Air Furnace.

Gases in this space not only heat the fourth side of the heating sections but also the air on the outside of the steel casing. Beyond this air space is the casing of the furnace. The figure shows return air entering from the side above the floor level. The damper at the top of the combustion space gives a direct path to the smoke pipe for use when fires are started. The great amount of heating surface available in this furnace makes it one of the most efficient of present-day warm air furnaces. The temperature of the flue gases will be lower, and one does not have to depend on heat from the chimney or smoke pipe to secure high over-all efficiency. The greater transfer of heat to the air enables the designer to get more directed heat to the rooms from the leaders and flues.

The circulating force from the hot air in these corrugated flues permits the installation of caps inside of the bonnet over certain flues to direct the hot air into long leaders or lines of high resistance.

A fan and filters for warm air heating are shown in Fig. 215. This is one of a type of air heating units built within a casing which is insulated and finished with Duco on metal or other attractive surface. Air passes to the fans through filters of large area and thence into the humidifying space at the lower part of the heating section. Water, heated by radiation from the combustion space, is cascaded over a series of dams and regulated so that there is little or no overflow to the sewer from the last

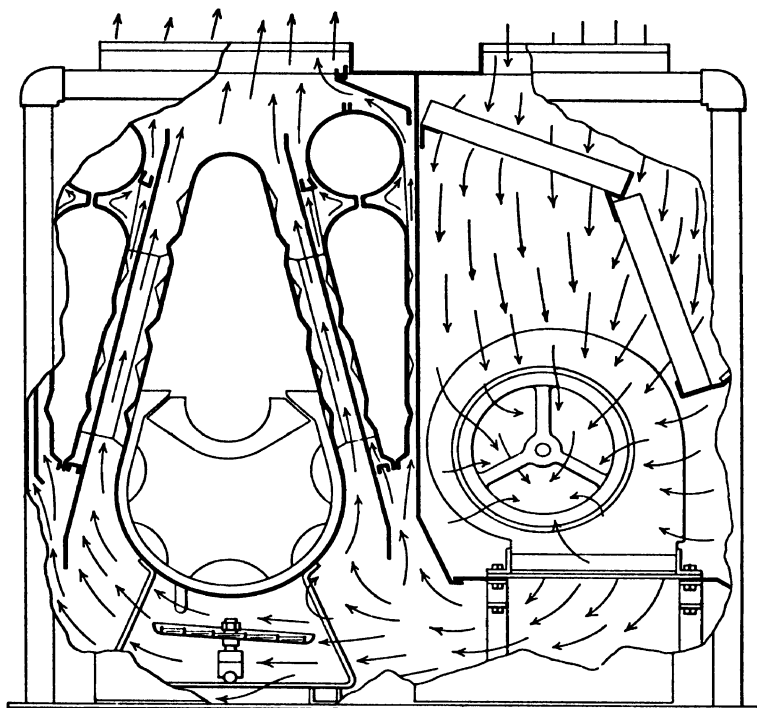


Courtesy of Kelsey Heating Co.

FIG. 214.—Warm Air Generator.

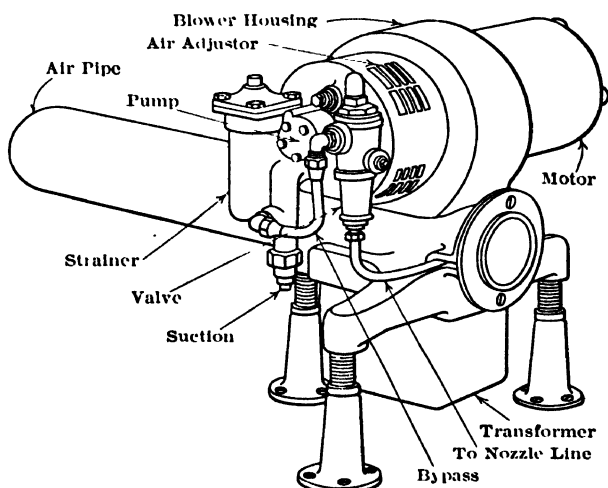
tray. After the air is properly humidified, it passes over the extended heating surface of five heat transfer sections and is delivered from the top of the casing to various leaders. Oil fuel is completely burned in the combustion chamber which is lined with firebrick and contains a refractory target against which the flame is delivered to aid combustion and maintain ignition. Hot gases pass from the back of the combustion chamber into the lower side chambers and from the front of these to the small upper passages and thence backward to the smoke pipe. The wipe sheets placed between the heating units absorb radiant energy and transmit this by conduction to the air while projections formed in the side sheets of the chambers cause eddies and increase the rate of heat transmission.

The oil burner, Fig. 216, consists of a motor-driven fan and rotary gear oil pump on the motor shaft, discharging a mixture of air and atomized oil from the end of the air pipe. The oil under pressure is



Courtesy of Delco Appliance Corporation

FIG. 215.—Oil-fired Warm Air Furnace.



Courtesy of Delco Appliance Corporation

FIG. 216.—Delco Oil Burner.

atomized as it is forced through the nozzle at a pressure of 100%. A check valve which opens when the pressure reaches 60% prevents oil from entering the nozzle unless the oil is at or above this pressure. The fan housing, air pipe and the base form a rugged unitary structure which carries the transformer for the high-voltage ignition spark at the end of the nozzle. A thermostat placed in the furnace flue or smoke pipe (called a pyrostat) is arranged to cut off the burner when the gas temperature falls below a given value as the result of improper combustion or the failure of ignition.

Certain warm air furnaces utilize gas as a fuel. They have an extended heating surface through which the products of combustion from the Bunsen burners are passed to the chimney flue. The gas supply and air supply are regulated by a thermostat in the building which actuates the electric motor valve to change the flow of gas. A gas pilot light for ignition is equipped with an electric device which prevents the admission of gas if the pilot is unlighted.

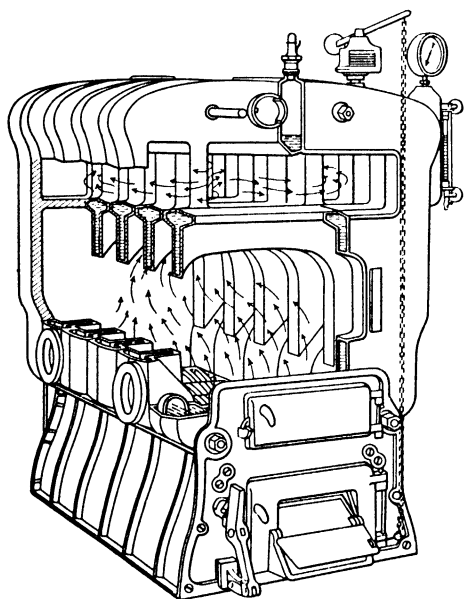


FIG. 217.—Ideal Sectional Boiler.

SECTIONAL HEATERS AND BOILERS

Sectional hot water heaters or sectional steam boilers are made of cast sections joined together by slip nipples. The unit is made up of a back section casting, a front section and several intermediate sections, the number being fixed by the desired capacity of the unit. The arrangement of the gas passages and shapes of sections vary with the manufacturer, but the general arrangement is shown in Fig. 217 which represents a

former type of the Ideal sectional boiler of the American Radiator Company. This figure represents a steam boiler with its water level just over the top of the upper gas passage. Were the same structure filled with water it would become a hot water heater and the water gage and the pressure regulator would be omitted. In this unit

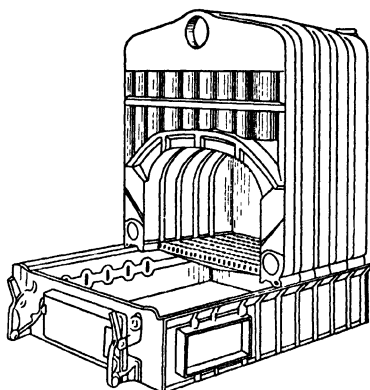
the cast sections touch, forming the combustion chamber and gas passages, but the back section is so cast that the products of combustion discharge into the central gas passage, and after traveling to the front they reach the end section, which is made so that the gases pass to the sides and thence to the rear where the streams unite at the outlet into the smoke flue. The front section contains an opening for firing, another for stoking and a top opening for cleaning the gas passages of soot. Sections of the form used at present by the American Radiator Company and other manufacturers contain many more gas passages arranged at two levels in place of the single level shown in the figure. A recent form of Ideal boiler is shown in Fig. 218.

The sections of Fig. 217 are united at three points by push or slip nipples, and four through-bolts hold these together. Water is introduced at the bottom of one or more sections, and steam (or hot water) is taken from the top of some of them. At times each section is connected to a collecting pipe above the boiler, and again steam may be taken from the faces of the end sections.

The sections of Fig. 217 are supported on a cast base which forms the ash pit. The shaking grate supports may be formed on this cast base or on the sides of the sections.

In this unit the firebox is deep so that a large charge of coal may be placed on the grate at one time. In fact, the grate of the smaller boilers or heaters will hold enough coal to last for 20 hr without refueling, and in the larger boilers the capacity is such that refueling occurs at 6 or 8 hr intervals.

The figure indicates a damper control which at low pressure opens a door in the front or side of the ash pit at the same time it opens the flue damper or closes the flue check draft door. When the pressure exceeds that for which the regulator is set, the draft in the flue is checked by opening the check door on the flue at the same time that the door on the ash pit is closed. These dampers and checks may be operated by thermostats in the building, and with hot water boilers, regulation is accomplished by thermostats not only in the building but also in the hot water outlet.

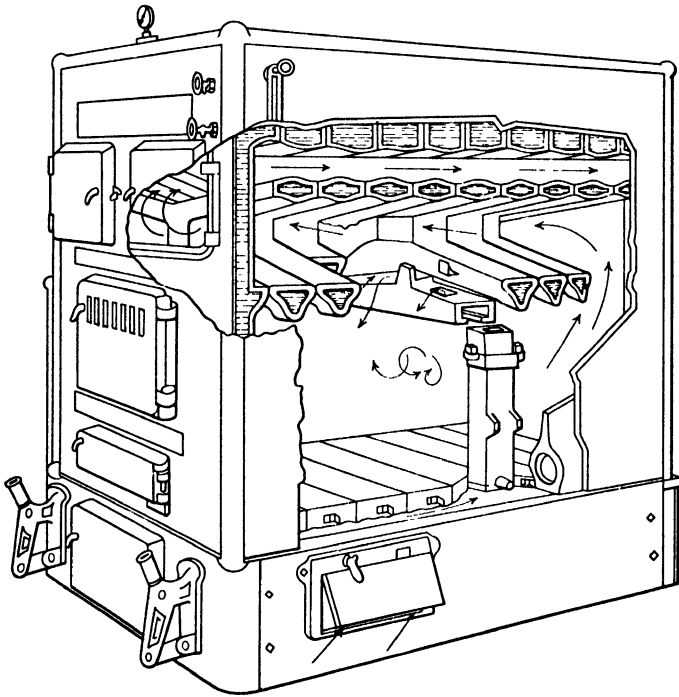


Courtesy of American Radiator Co.

FIG. 218.—Ideal Sectional Boiler.

The boilers should be covered with proper heat insulation, and on present-day boilers, insulation within metal jackets of attractive finish is furnished by the manufacturer. One further step is to place the boiler or heater within an enameled metal cabinet which covers the firing and ash-pit doors, making it necessary to open the large front door when refueling or ash removal is required.

For the smokeless burning of soft coal a supplementary supply of hot air is introduced into the volatile gases at the rear of the grate. In



Courtesy of Crane Co.

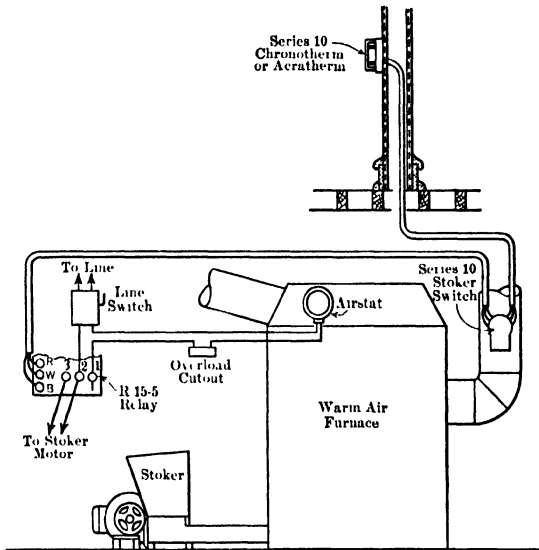
FIG. 219.—Insulated Sectional Boiler with Smokeless Attachment.

the form of furnace, Fig. 219, used by the Crane Company, two risers near the back of the grate extend through the fuel bed from the ash pit to a distributing arch or oxidizer at the top of the combustion chamber, and air from the ash pit which enters from the top of this arch has been so heated that it ignites the unburned volatile matter as this passes to the rear of the combustion chamber.

This figure indicates the metal casing finish which is held by edge corner strips supported by four corner cups which may be released by taking out four screws, thus freeing the sheet-metal covers of the five

insulating slabs. The draft control door is shown on the side of the ash pit.

Boilers or hot water heaters containing coal magazines for increasing the period of time between stokings are built by several manufacturers, and units are also built for oil and gas fuel. With oil, the combustion chamber is made the full depth of the setting and a large furnace volume is necessary for complete combustion of the oil before chilling the products of combustion. In addition, the gas path is lengthened to insure the full removal of heat. When gas fuel is employed, one manu-



Courtesy of Minneapolis Honeywell Regulator Co.

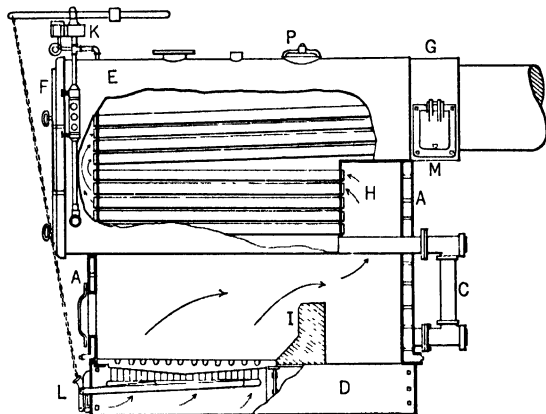
FIG. 220.—Stoker Fired Warm Air Furnace.

facturer, to absorb the heat, uses cast sections containing a number of pin points similar to those on the pin-point radiators.

A number of mechanical stokers for sectional boilers and warm air furnaces, as well as for larger boilers, are available today. Fig. 220 illustrates one of these applied to a small furnace. Coal from the hopper is fed by a screw to the underfeed grate, and a fan blower supplies the necessary air for combustion. By controlling the driving motors by a thermostat the feed and rate of combustion are varied with the heat demand. Coal is placed in the hopper by hand or sometimes is fed to it from the coal bin by a screw conveyor, and even ashes may be removed from the ash pit to ash cans by such a conveyor.

A number of forms of sheet-steel low-pressure heating boilers are in

use, of which that of the Pacific Steel Boiler Corporation, Fig. 221, is selected as an illustration. In this the firebox is made up of four water leg walls, *A*, containing the necessary stay bolts. The firebox is a single closed unitary structure joined to the separate shell of the boiler by a number of short pipe connections on the side and by a large circulating pipe, *C*, at the rear. The side connectors are shown at *Z* in Table A-28. The firebox rests on a base, *D*, which supports the shaking grate and forms the ash pit. The cylindrical boiler shell, *E*, rests in the saddle formed by the water legs and contains two nests of tubes, the lower nest leading the gases from the combustion chamber to the front of the shell at which point the smokebox, *F*, returns them to the upper nest of tubes



Courtesy of Pacific Steel Boiler Corporation

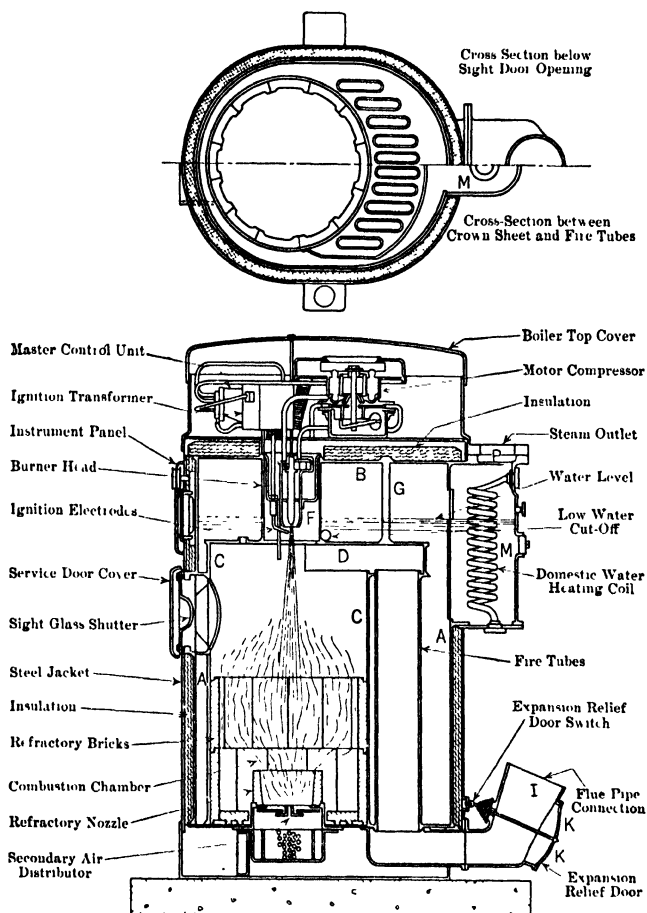
FIG. 221.—Welded Sheet Steel Boiler.

through which they pass to the rear smokebox, *G*. Half of the shell is extended out to this point, the lower part of this extension forming the top of the combustion chamber *H*. The figure includes the bridge wall, *I*, at the rear of the grate, as well as the water column, the pressure regulator, *K*, which raises and lowers the ash-pit door, *L*, and the draft check, *M*. The steam gage is shown and the outlet for steam as well as that for the attachment of the safety valve are seen near the man-hole, *P*. The grate is of the shaking type.

These low-pressure boilers are fabricated from sheet steel by welding, some using the gas flame, others the electric arc. When this welding is done in accordance with the A.S.M.E. Boiler Code, a safe structure results, as is indicated by the large number of welded heating boilers in use.

Another successful welded boiler using oil as a fuel is that fabricated by the General Electric Company.

This boiler, Fig. 222, is built up of plates arc-welded together and consists of a shell, *A*, with a top-head sheet, *B*, and a cylindrical combustion chamber extending through the bottom sheet and made up of



Courtesy of General Electric Co.

FIG. 222.—G. E. Heating Boiler.

side sheet, *C*, and crown sheet, *D*. At the top of the combustion chamber, a cylindrical chamber extending from the crown sheet to the top sheet is formed from a tube by welding. The chamber receives the burner head. This chamber, *F*, also acts as a central stay to support the crown sheet and top head. Stay bolts marked *G* are welded to these

sheets of the boiler. The crown sheet is continued beyond the combustion chamber to form part of the smokebox to which is connected a series of fire tubes extending through the bottom head of the boiler.

The side sheets of the boiler shell and the top head are insulated by a non-conducting cover. On top of the boiler is mounted a compressor motor which forces oil and atomizing air through the burner and delivers a cloud of oil particles downward with the primary air which enters around the burner. The slow burning of part of the oil associated with primary air warms the mixture so that when it meets the up current of the secondary air it burns quietly in the furnace lined with refractory material. The oil distributed from the atomizer is ignited by the electrodes at the nozzle which receive high-voltage current from an ignition transformer. Much of the heat from the combustion of the oil is transmitted by radiation from the firebrick and the burning oil particles, and in addition to this, the products of combustion as they pass upward through the combustion chamber and downward through the fire tubes arranged around it, deliver heat to the water of the boiler, and the products of combustion pass out through the flue connection, *I*, at a low temperature. An expansion relief door, *K*, opens under excessive temperature to admit air.

An extension to the shell of the boiler at *M* holds a device which heats water for domestic service by circulating it in a coil surrounded by hot boiler water. Steam is taken from the boiler at the outlet, *P*.

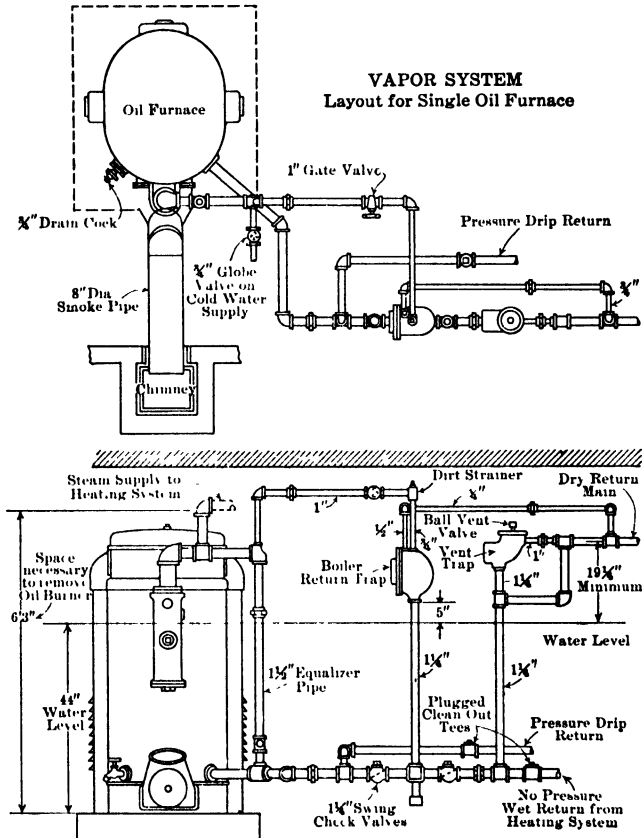
The supply of hot water at 140° F to 180° F, for domestic use or for use by occupants of a building, is often cared for by separate gas, oil or coal-fired water heaters, or the water may be heated by a coil in the main boiler or warm air furnace. At times steam may be taken from the system into the coil of a water tank to heat this water. The tank operates as a reservoir to meet excessive rates of use of short duration, building up the supply during the period of slight demand. For general use, including baths, 40 gal of hot water should be allowed per person per day as probable load, and for lavatories only 5 gal per person per day may be used. The tanks for equalizing the demand in large apartments with baths should have a capacity of 10 gal per person, while the capacity for residences should be 20 gal per occupant. The A.S.H.V.E. Guide for 1936 recommends about 30 gal of tank capacity for each bath equipment. For office buildings the tank capacity would be 2½ gal per occupant.

The amount of heat needed to increase the temperature of 1 gal of water 100° F is 835 Btu, which is 280 Btu per hour if the heating in the tank is done in three hours. If a properly insulated tank is used in this way, the heating load put on the boiler will be the equivalent of

1.15 sq ft of E.D.R. per gal of volume. This equivalent surface must be added to the heating surface of the building to fix boiler capacity, or it is used to select an independent water heater.

The domestic water heating coil of the G.E. boiler is the equivalent of $52\frac{1}{2}$ sq ft E.D.R. (steam).

The complete boiler and insulation is covered with a steel jacket which is provided with a service door, including a sight glass shutter



Courtesy of General Electric Co.

FIG. 223.—Installation Details for G. E. Boiler.

for visual inspection of the flame, and the compressor with its control mechanism is entirely included within a metal cover placed over the top of the unit.

The boiler is under the control of a thermostat in one of the occupied rooms, which regulates the fire to maintain the necessary temperature. A master control unit on top of the boiler operates in conjunction with the room thermostat and is so connected with the flame chamber instru-

ment that it operates to regulate the compressor motor, the oil valve and the ignition discharge. The master control also operates in response to the requirements for domestic hot water, opening up the oil supply and ignition system when the temperature of the water falls. It also operates to shut off the oil supply if the steam pressure is too high or whenever the system fails to ignite the oil. Normally, the water level in the boiler is maintained by the return, and should this water reach a low level, the oil is shut off automatically. The cross-section of the boiler shown in the figure indicates the arrangement of the fire tubes which receive the gases from the top of the combustion chamber and deliver them into the smoke pipe.

In Fig. 223 the arrangement of the piping for a single oil furnace is shown. Several furnaces may be combined in parallel to care for larger installations; in one of these, at Mount Holyoke College, 120 units are applied in one power house. The two sizes of units, 30.5 and 53.5 sq ft. of heating surface, have capacities of 133 and 275 Mbh.

When much steam at high pressure is needed power boilers must be used and such are described in the author's textbook, "The Elements of Power Generation." For the design and selection of such boilers, reference is made to Gebhardt's "Steam Power Plant Engineering."

Selection and Installation of Furnaces, Heaters or Boilers. Chimneys. The selections of units, similar to those described in this chapter, are made from the heat requirements of the building or group of buildings and the accepted ratings of units by the manufacturer, although from the known efficiencies of these generators at given outputs and the rate at which fuel may be burned, the size of the grate or the number of burners and the size of the combustion space may be computed and checked against the manufacturer's tables.

$$\frac{\text{Heat requirement of system per hour}}{\text{Efficiency of apparatus}} = \frac{\text{Energy needed in fuel per hour}}{\text{hour}} \quad (12-1)$$

$$\frac{\text{Energy needed per hour}}{\text{Heat value of 1 lb or 1 cu ft of fuel}} = \frac{\text{Pounds or cubic feet of fuel per hour}}{\text{per hour}}$$

$$\frac{\text{Pounds of fuel per hour}}{\text{Rate of combustion in lb per sq ft per hr}} = \frac{\text{Total grate area in square feet}}{\text{feet}} \quad (12-2)$$

Rate of combustion for

Warm air furnaces, 2 to 4 lb per hr per sq ft

Small boilers, 2 to 4 " " " " " "

Larger boilers, 10 to 40 " " " " " "

$$\text{No. of units} = \frac{\text{Total grate area}}{\text{Grate area per unit}} \quad (12-3)$$

For oil or gas

$$\frac{\text{Pounds or cubic feet of fuel per hour}}{\text{Capacity of burner in pounds or cubic feet per hour}} = \text{Number of burners} \quad (12-4)$$

A given amount of draft must be available from the chimney to draw the products of combustion through the generator and deliver them at the chimney top, and also with most fuels this draft must be sufficient to draw the air for combustion into the firebox through the fuel.

This draft is a function of the chimney height, the temperature of the outside air and that of the flue gases, while the resistance to flow depends on the resistances of the grate and fuel, the gas passages of the generator, the flue and the chimney. The sizes of smoke flue, chimney flue and the minimum height of chimney are given by some of the manufacturers, and the A.S.H.V.E. in their code, Minimum Requirements for the Heating and Ventilating of Buildings, gives Table A-30.

When the size of an installation is larger than that given in the table and power boilers are required, the resistances to flue gas flow are computed, after which the chimney height is determined to give this draft with the highest outside temperature and the lowest chimney temperature to be expected at the time of the greatest coal consumption. The area of the chimney flue is then determined as is discussed in books on power plant practice.*

Small chimneys are usually built as a part of the wall of the building, or within the building, and are constructed with a tile flue lining around which are walls of at least 4 in. of brick work. The Underwriters rules require that chimneys extend 3 ft above the roofs of the buildings, and

* The draft must produce the small initial velocity head of the entering air.

$$x \left(\frac{V}{4005} \right)^2 = h_1$$

The resistance at the grate in inches of water, as given by Kidwell, is shown in Fig. 224.

The resistance through the boiler may be taken as

$$0.15 \left(\frac{\text{Evaporation per square foot of surface}}{3.0} \right)^2 \text{ in inches of water.}$$

The resistance of a bend in the flue is usually taken as 0.4 of the velocity head although some assume each right-angle bend to require 0.05 in. of water.

The resistance of the smoke pipe or breaching and that of the chimney flue is computed by the methods of Chapter VIII applying the necessary factor for the density of the hot gas which can be considered as air as its composition in general has an average molecular weight equal to that of air. With gases at 540° F and at

2 ft above any adjacent ridges, parapet walls or other wind obstructions. The author prefers to increase these limits by 2 ft. The flue should be straight, if possible, and all joints in lining and brick work should be filled with mortar. Any change of direction must be made gradual to

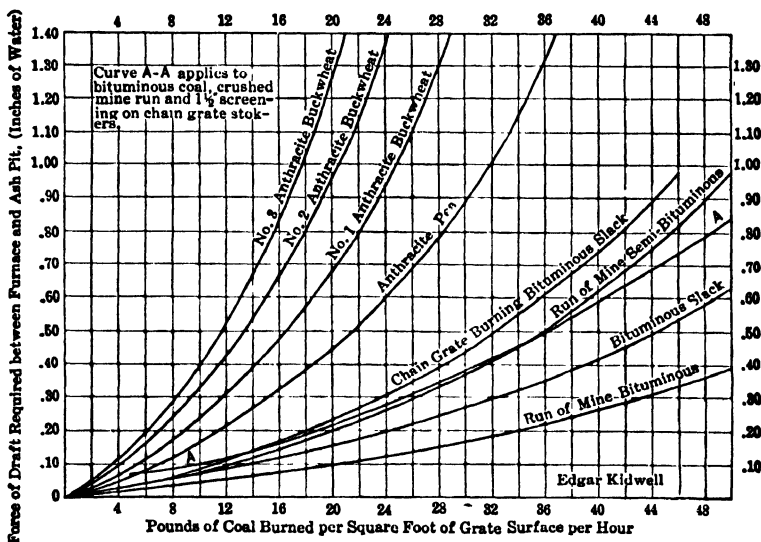
29.92 in. of mercury, Gebhardt computes the draft loss in a brick-lined stack or flue to be

$$\text{Inches of water} = 8 \times 10^{-6} \times \frac{LV^2}{D}$$

L = length in feet

V = velocity in feet per second

D = diameter in feet



From *Steam Power Plant Engineering* by Gebhardt

FIG. 224.—Draft Requirements at Grates.

A common velocity to use is 30 fps, which, with 540° F, standard barometric pressure and a length of 100 ft, would reduce the above to

$$\text{Loss in brick-lined flue per 100 ft} = \frac{0.72}{D} \text{ in. water}$$

From the sum of the friction losses for all parts of the system, the height of chimney, L , required for this draft, with an allowance of 20%, is found under standard atmospheric pressure by:

$$\text{Draft in inches of water} = 7.65 \left[\frac{1}{T_a} - \frac{1}{T_c} \right] L \times 0.80$$

T_a = absolute temperature of atmosphere in F degrees at time of greatest draft demand;

reduce resistance. The top of the chimney is usually fitted with a cap of stone, concrete or metal, and at the bottom of the chimney there should be a chamber with a cleanout door extending below the inlet of the smoke flue. Smoke flues leading to small chimneys are made of galvanized iron of No. 18 thickness.

$T_c = \frac{3}{4}$ of absolute temperature of entering flue gases in F degrees to care for cooling;

L = chimney height in feet.

With the height of chimney known, the cross-sectional area of the chimney passage is given by the formula of Wm. Kent:

$$\text{Boiler horsepower developed} = \frac{\text{Heat generated in Mbh}}{33.472}$$

$$\text{Effective area of gas passage} = \frac{\text{Boiler hp}}{3.33 \sqrt{L}} \quad (12-5)$$

Effective area = area of flue less a 2 in. strip around circumferences.

To obtain the velocity at any point the quantity of gas and its temperature must be known.

The heat needed by a given plant is q Mbh, and this would require an amount of fuel equal to

$$\text{Fuel per hour} = \frac{1000q}{\text{Eff} \times q_f}$$

$$q_f = \text{Btu per pound of fuel}$$

It is shown by Gebhardt that the pounds of air required to burn a pound of fuel is approximately equal to $\frac{7.5q_f}{10,000}$, in which q_f = heat value of the fuel in Btu per pound, and if 40% excess air is used to prevent the formation of carbon monoxide the amount of gases in the products of combustion is given by

$$\text{Lb per hr} = \frac{1000q}{q_f \text{ Eff}} \times \left(\frac{10.5q_f}{10,000} + 1 \right) = W$$

By the gas formula the volume of this weight at any temperature may be found by

$$\text{Volume} = \frac{WRT}{p}$$

From the volume the cross-sectional area could be found at any particular temperature of gas and for a desired velocity.

$$A = \frac{\text{Volume}}{V}$$

The temperature of flue gases will vary from 400° F to 700° F, and in finding velocities this must be assumed.

For large chimneys, the inner lining of the core is made of firebrick to a height of 50 or 70 ft above the grate, and sometimes even for 100 ft. Even here the chimney should be part of the building wall, or within the building if this is possible. In such cases the shell of the chimney and the separate core are designed to support their weight by direct compression. Whenever the chimney is independent of any other structure its outer shell must be designed to resist the stresses due to the wind as well as those due to the weight of its parts above any level at which it is investigated for stress, while the core carries merely its own weight above such levels.

Tables A-28 to A-32 give manufacturers data for a few generators, and the designer should have the data sheets for the complete line of products of a number of manufacturers for use in selecting units to suit his designs for different installations.

Warm Air Furnaces. To select the warm air furnace for the building discussed in Chapter XI, for which the heat supply is 115 Mbh and the combined area of the leaders is 832 sq in., reference is made to Table A-32. The data of the Mueller full front furnace show that F-54B is too small, and so No. 390 will have to be used. This will care for 975 sq in. of leaders and 132 6 Mbh. The grate area is 605 sq in. and the diameter of smoke pipe is 10 in.

If anthracite coal of 12,500 Btu per lb is used with an efficiency 60% and a rate of combustion of 4 lb per sq ft per hr, the required grate area would be

$$\text{Grate area} = \frac{115.0 \times 144}{12.5 \times 0.60 \times 4} = 553 \text{ sq in.}$$

This result also indicates that F-54B is too small although No. 390 is over size. It must be used if there is no intermediate size.

In the same manner the Kelsey generator may be selected and checked. Size 27 will be required. A 9 in. smoke pipe and a 12 in. \times 12 in. chimney flue are to be used with this. The face of the grille for return duct should be 30 in. \times 36 in.

Steam Boilers and Water Heaters. In selecting boilers and heaters the square feet of radiation of the system includes water heating and any other service. The code requires this allowance be used, as these devices are rated on this unit of capacity. Some engineers add 15% to 25% to the square feet of radiation or to the Mbh to allow for the heat loss from pipe. The author does not follow this practice, as this heat flow which actually occurs from pipes diminishes the amount to be supplied by the radiators if the computed heat demand had been supplied by the radiators alone. The addition of this 15% does give a reserve of generator capacity. The A.S.H.V.E. Code requires an additional amount of capacity to supply the steam needed in warming up. This "pick up" amounts to 65% of the regular demand in small boilers and 40% in large ones. When oil is employed this increase is 25%.

For the steam or water heating of Chapters VI and VII, Table A-29 indicates the size of Ideal water tube boilers to be used. It is seen that for 50% pickup 178 Mbh capacity will be needed. S or W-2309 will care for 180 Mbh with an efficiency of 73.5% and S or W-2310 would have an efficiency of 74.0% because of the lower combustion rate. The fuel would last 13.8 hr between full firings with

the second unit. The rate of combustion is 3.3 lb per sq ft per hr, the draft required is 0.04 in. of water and the chimney should be 40 ft high with a 12 in. \times 12 in. lining. This unit will care for 750 sq ft of E.D.R. with steam and 1200 sq ft with water as the heat demand is the same for each system. The dimensions of these units are given in the figures of Table A-31.

The Pacific steel heating boilers are not intended for small buildings, and to illustrate the selection of such it is assumed that with all extra allowances the capacity is 14,000 sq ft of E.D.R. The heat demand is 3360 Mbh. Three units will be installed, each of capacity 5000 sq ft E.D.R., as, if one were under repair, the two would carry the load, the allowance being cared for at this time by forcing the two boilers. This, from Table A-28, would require a boiler No. 2333. The grate area of 15.4 sq ft requires a combustion rate of 9 lb of coal per sq ft per hr for which the chimney must be of some height.

$$\text{Grate area} = \frac{5000 \times 240}{12,500 \times 0.70 \times 9} = 14.7 \text{ sq ft}$$

In general, the data sheets of the manufacturers are based on actual tests of their products, so that the given capacities and performances may be relied upon by the engineer. However, it is well always to check the grate area and especially the furnace volume for the heat release per hour, as for complete combustion there must be a proper volume in which the fuel burns before passing into the small spaces between heating surfaces.

The following table prepared by E. G. Bailey gives the allowable Mbh to be liberated per cubic foot of furnace volume.*

Fuel and Firing	Per Cent Excess Air	Mbh per Cu Ft Furnace Volume
Wood refuse.....	60	15
Coal, natural draft, chain grates... ..	54	25
Coal, overfeed stoker....	53	30
Coal, hand fired.....	53	40
Coal, forced draft, chain grate....	48	25
Coal, forced draft, underfeed...	45	30
Coal, pulverized.....	27	11
Oil, steam atomized.....	23	22
Oil, mechanical atomized..	22	31
Gas.....	20	21

PROBLEMS

1. Select the number and horsepower of the boilers at 250% rating to be installed in a boiler house to supply steam for heating buildings with a total of 1,000,000 sq ft

* Mechanical Engineering, July, 1926.

of E.D.R. What would be the total grate area for this plant? Would an extra boiler be added to the number required?

2. Select the warm air furnace to be used for the building of problem 1, Chapter XI. What would be the size of the smoke flue? What would be the size of the chimney?

3. Select the Ideal boilers to supply 600 Mbh to the radiators of a building for steam heating. What are the over-all dimensions of the units? What sizes of grate, smoke flue and chimney would be used? What size Ideal water heater would be chosen for this plant if the system were for hot water in place of steam? Select the Pacific welded boiler for this service and give leading dimensions. How many General Electric boilers would be needed for this service? Add: 10% for piping. Equivalent surface for 30 bath fixtures. 50% pick up.

CHAPTER XIII

AUTOMATIC CONTROLS

The control of temperature, humidity, air, velocity, steam pressure, draft, oil feed or any operative condition is important for the comfort of the occupants of a building and for the economical operation of its heating or air-conditioning plant. The desirability of maintaining the temperature of a building or space within the comfort zone is quite evident, and when it is realized that the cost of heating depends on the difference between the temperatures of the rooms and that of the outside, it is seen that when the temperature of any room is above that required for comfort the heat demand is greater than the designed quantity. This means a loss in efficiency. There is need for a device to maintain temperature in a heated room or regulate it to within a few degrees of some desired value. A number of systems have been devised, some of which operate in single rooms to open and close radiator valves or to regulate the mixing dampers of the indirect system; others operate on the draft controls of the boiler, hot water heater or hot air furnace, and others regulate the pressure of the steam supply to the system or to one of its zones. In this way the amount of heat supply to any one part of the system is just equal to that required for proper performance and waste does not occur. Many of these systems contain delicate parts which may be disarranged by occupants of the building who do not understand the device and in attempting repairs or adjustments injure the parts. Such devices must not be handled by persons unfamiliar with their construction.

The purpose of all these systems is economy as well as comfort, and to compare the cost of heating from year to year, this is reduced to the cost of some unit. The degree day is used as this unit. The degree days for any month are found by adding together the difference between the average building temperature and average outside temperature for each day of the month. The sum of the degree days for the months of the heating season or for the year is the total degree days for that period. Although 70° F is a common inside temperature for most buildings in the United States, Walker has found that for Detroit degree days figured from 66° F as the inside temperature give values which are in better agreement with actual heating costs than those computed from actual

temperatures. The American Gas Association uses 65° F as the point from which to measure the unit.

Although the degree days should be computed by adding the degree differences each day, a close approximation may be obtained by the use of the monthly average temperature. Thus for a certain city the average monthly temperature is subtracted from 65° F, and when this difference is multiplied by the days per month the degree days for the month are obtained, and from these the degree days per season.

January.....	$31 \times 31.8 = 986$	August.....	$31 \times$
February.....	$28 \times 46.1 = 1292$	September.....	$30 \times$
March.....	$31 \times 38.8 = 893$	October.....	$31 \times 11.9 = 369$
April.....	$30 \times 16.5 = 495$	November.....	$30 \times 17.2 = 516$
May.....	$31 \times 2.5 = 77$	December.....	$31 \times 32.8 = 1017$
June.....	$30 \times$		
July.....	$31 \times$	Total.....	<hr/> = 5645

The temperature base from which degree days should be computed for any given building or system may be found by plotting the heat used or pounds of heating steam per day against the outside temperature. These points lie along a straight line in general, and the point at which this line cuts the axis of temperature fixes the temperature base.

Degree days unfortunately do not include the velocity effect on the outside air films and on the window leakage, so that the unit cannot be used to compare the operations of plants in different localities with accuracy. It may be used to compare installations in the same locality and to compare the operation of the same plant in various years as these wind conditions are not so variable from year to year. To compare different plants in the same locality, the yearly amount of steam, fuel, electric heat energy or heat energy is reduced to the yearly amount per degree day per cubic foot of heated volume, per square foot of heated floor space, or per square foot of equivalent direct radiation. The first two of these (per cubic foot of heated volume or square foot of floor space) vary with the type of construction so that similar buildings only may be compared. The latter method of comparison depends on the method of design, the peculiarities of the designer and the desired temperatures of the rooms, all of which fix the amount of radiation. Although it is difficult to make comparison of different plants, the results from similar buildings may be helpful in judging the operative efficiency of a given plant.

By determining the pounds of steam or Mbh per degree day per 1000 cu ft of space heated under operating conditions with automatic control equipment and the performance without these devices, the great saving they effect can be estimated.

Orifice Control. With main risers for branch mains of tall buildings or with steam mains below ground when steam is supplied to a building from a central heating plant (called district heating) it is customary to carry high-pressure steam in these mains and to reduce this pressure to that needed for the branch main or the building through a reducing pressure valve. With high-pressure steam a small pipe may be used for a given weight of steam per hour, and moreover there is less heat loss from the covering of the small pipe with higher temperature difference. Reducing pressure valves are of various forms; that shown in Fig. 225 illustrates the general principles of action. High-pressure steam at point *A* passes the balanced double beat throttle valve *V* and is delivered at *O* at a lower pressure. The pressure at *O* acts on diaphragm *D* against a force from weights *C* and *T* on a lever *L* which opens the valve, so that when the force exerted on the diaphragm by the steam pressure from *O* is greater than that from the weights the valve closes enough to keep the low pressure at outlet of the desired value. When exhaust steam is used for heating it is customary to add to the system a so-called back-pressure valve, which is merely a safety valve releasing steam to the atmosphere whenever the pressure exceeds the maximum allowable back pressure on the exhaust.

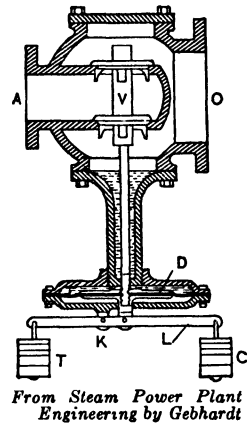


FIG. 225.—Reducing Pressure Valve.

In heating systems equipped with return trap valves on the radiators, steam is never discharged to the return but by manipulating the modulating valve the full area of the radiator may be heated even in mild weather, the excess heat liberated to the room being removed by an open window. If the system is properly piped and supplied with steam, all radiators will receive their needed steam even though some or all of them are being fully heated. To cut down the waste of steam in mild weather by overheating some radiators it has been proposed to alter the pressure difference between the supply and return at such times so that the steam used by any one radiator cannot be excessive. Of course, with this low pressure in mild weather the radiators with wide-open valves can rob other radiators as the total flow of steam has been reduced, and although the radiator cannot condense more than $\frac{1}{4}$ lb of steam per hr per sq ft, this amount would mean a waste for such radiators at such times as well as an interference with the proper heating of the other units. To arrange a system which prevents this waste,

metering orifices, Fig. 226, are placed in the inlet nipples of each radiator through which only a definite amount of steam can pass for a given difference of pressure between that of the steam main and the exhaust main. This smaller amount of steam will be completely condensed by part of the radiator surface, and only water reaches the return trap valve, which remains open, bringing the pressure within the radiator to that of the return main.

The area, a , of the orifice in thin brass plates, 0.02 in. in thickness, for a given radiator surface A may be computed for the coldest day with full steam pressure difference by a modified Grashof formula for discharge of steam through orifices as shown by Sanford and Springer.*

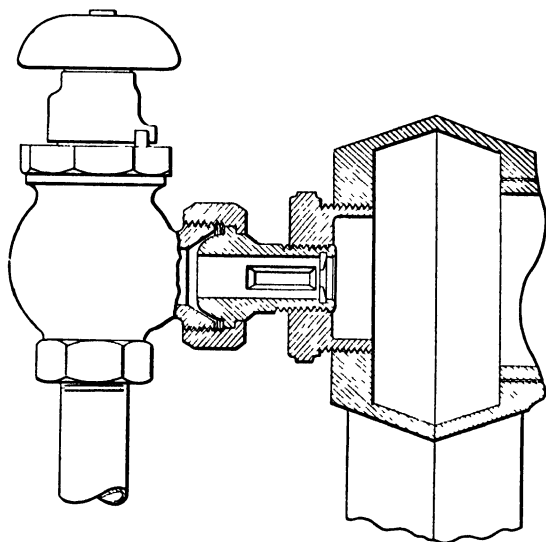


FIG. 226.—Webster Metering Orifice.

$$a = \frac{A}{240kp^{0.97}};$$

a = area of orifice in square inches;

A = E.D.R. of radiator in square feet;

p = steam pressure at valve in pounds per square inch;

k = constant depending on ratio of p_b/p ;

p_b = pressure in return line at radiator in pounds per square inch.

The value of k is plotted in Fig. 227 for different values of the ratio p_b/p . (From results of Sanford and Springer.)

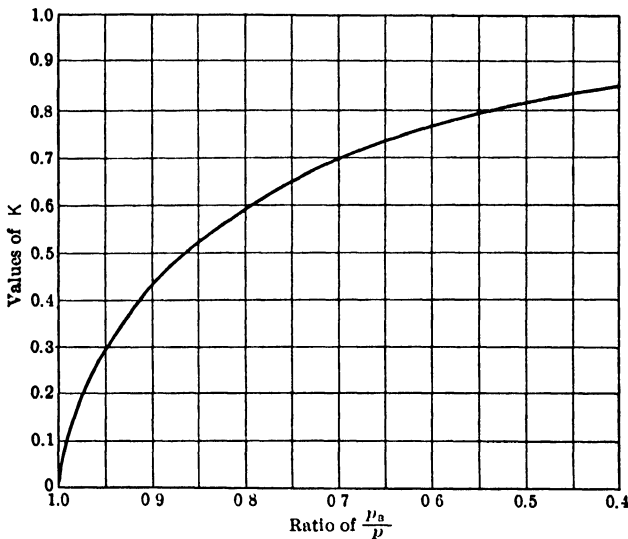
* Flow of Steam through Orifices into Radiators, by S. S. Sanford and C. B. Springer, A.S.H.V.E., Trans., Vol. 37, 1931, p. 371.

With this value of a known for a given radiator, the supply pressure needed with a given value of p_b for any fractional part C , of the full heat from a radiator, $240A$, can be found by successive approximations, using the expression

$$p^{0.97} = \frac{CA}{240ak}$$

C = portion of heating value of radiator surface needed in milder weather.

From this equation the pressures on each side of the orifice may be found by trial if both p and p_b vary, and if the system is maintained with these pressures no radiator can receive enough steam for its complete



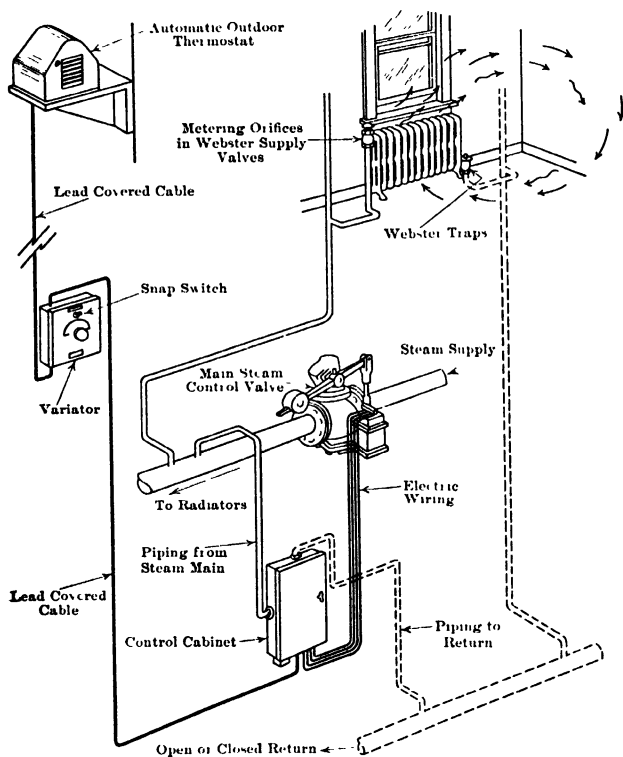
From Trans. A.S.H.V.E., Vol. 37, 1931, by Permission

FIG. 227.—Coefficient for Orifice Discharge.

heating, and the heat given to any room is limited by these orifices and the maintained pressure difference in the building. Moreover, as long as the desired steam pressure is carried throughout the system, each radiator will receive its allotted amount of steam if the valve is wide open.

The operating devices for one such method of temperature control (Webster system) are shown in Fig. 228. Steam, at any pressure, is brought to the control valve of a reducing pressure type which is regulated by an electric motor to maintain a definite difference between the pressure in the main conveying steam to the radiators and the pressure

in the return. The control cabinet contains a mercury contact tube, one end of which is under radiator supply pressure, the other, return pressure. The pressure difference at which contact is made to start the motor for opening or reverse it for closing the main supply valve is fixed by hand or by an atmospheric thermostat, the main valve being closed when the outside temperature is such that no heat is needed (70°F



Courtesy of Warren Webster and Co.

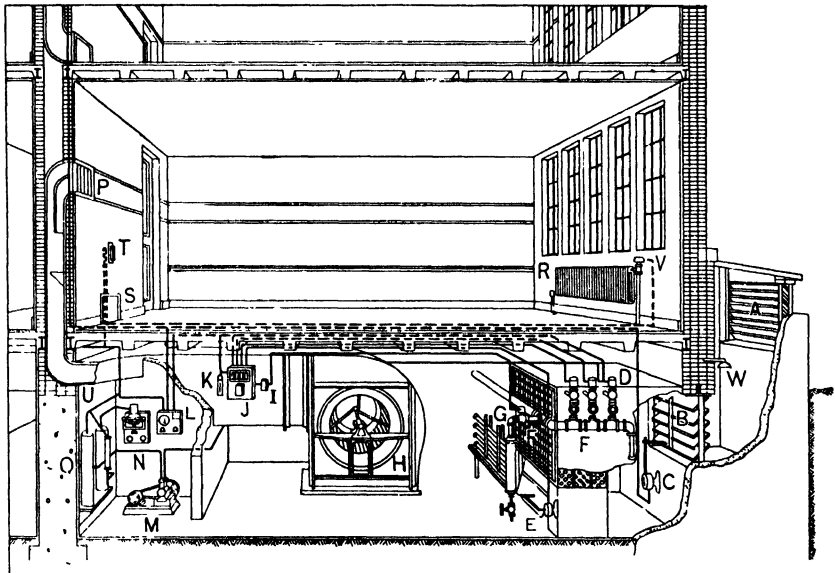
FIG. 228.—Webster Moderator Control.

possibly), and at minimum temperature it is opened the amount to give the full pressure in the mains for which the system is designed. A variator permits the manual control of the desired pressure difference for extra heating in the morning as well as the complete closure of the control valve. For large installations Warren Webster and Company, as well as others manufacturing such equipment, propose the division of the system into different zones so that each zone may be regulated

for the special conditions of temperature of the rooms associated with it or for the sun and wind effect at different times during the day.

The application of the orifice system with its variations of pressure to suit weather conditions has so reduced the waste of steam in times of reduced heat demand that savings of over 15 per cent in cost of heating for a complete season have been reported.

The Johnson system of temperature control has been used for many years. In it thermostats *T*, Fig. 229, regulate the flow of compressed air from the storage tanks *O* to the motor ends of the radiator valves *V* to maintain the temperature of the room. This air is put into the tanks



Courtesy of Johnson Service Co.

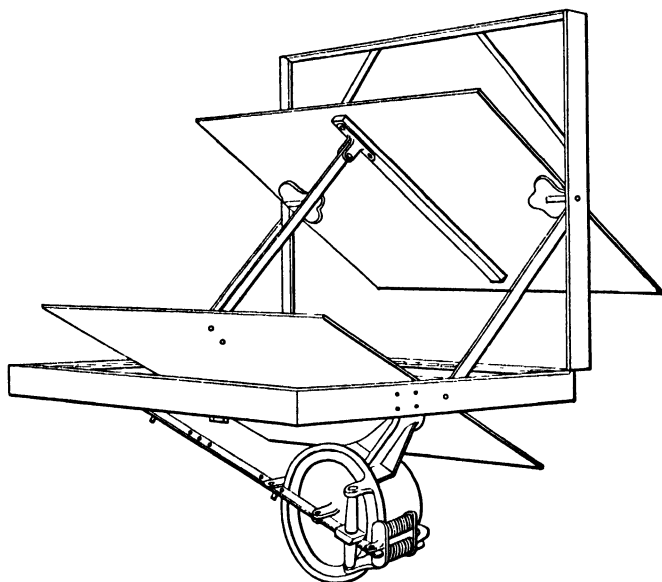
FIG. 229.—Johnson System of Temperature Control.

by the motor-driven compressor *M* which is started by the governor switch *N* when the air pressure drops to 18% from use in the valve heads or other motors. (Motor is shut off when 20% is reached.) The air for compression is taken through a strainer *W* in the cold air flue at the right of the figure. The pressure in the discharge line is maintained at some desired value, about 15%, by the reducing pressure valve *U*.

The figure illustrates a split system of heating, ventilating and air conditioning as the radiators *R* care for the heat losses while the indirect radiators *F* temper the air delivered through grille *P* in the desired condition of humidity and temperature. Air for this purpose is drawn from the outside through the louvres *A* and the damper *B* which is

under the control of the pneumatic switch of the sylphon damper motor *C*. When the damper *B* is partially closed another damper leading into this space in front of the radiators *F* from the return duct is opened partially to give the desired amount of recirculated air. The outlet for return air from the rooms is at *S*.

The multiple compound insertion thermostat *J* extending across the discharge duct of the fan *H* maintains the proper temperature of the ventilating air by regulating the supply of compressed air to the sylphon motor heads of the steam valves *D* on the indirect heater coils and to the motor operating the by-pass damper *E*, while the humidostat *I*



Courtesy of Johnson Service Co

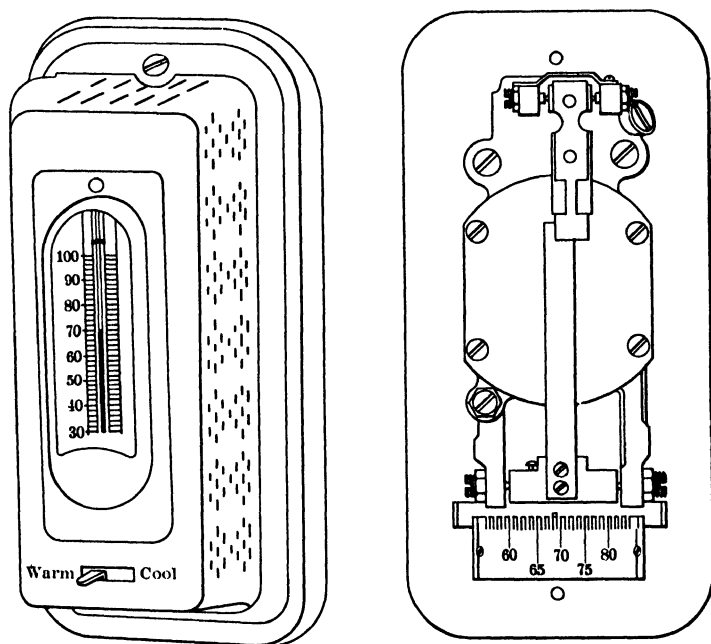
FIG. 230.—Dampers and Air Motor.

regulates the humidifier *G* to give the proper moisture content by evaporating water sprayed over a steam coil. Thus air of proper temperature and humidity enters the suction chamber in front of the fan.

Similar devices would be used for a complete indirect plenum system, a unit system or a complete direct system. The thermostats, humidostats and even devices to regulate static pressure or duct velocity control the flow of compressed actuating air at 15% pressure to the various motors which move valves or dampers.

The valve motors were illustrated in Fig. 75, and in Fig. 230 a sylphon damper motor is shown operating the mixing dampers on a hot and tempered air supply of a single duct system.

The Johnson Intermediate Standard Thermostat is shown in Fig. 231 and that for supersensitive action is shown in Fig. 232. Air at 15# gage pressure enters at point *EN* of the body *A*, Fig. 232, and passes through the body of the thermostat into the space around the main valve *D* and the pin valve *H*. This pin valve is slightly open so that air leaks into space to the right of the right-hand diaphragm *W*, and as this space is connected to the passage behind the leak port *B* the air will be discharged if the port is open. In the figure this leak port is closed by the seat *P* on the strap or holder *HH* pivoted at the top.



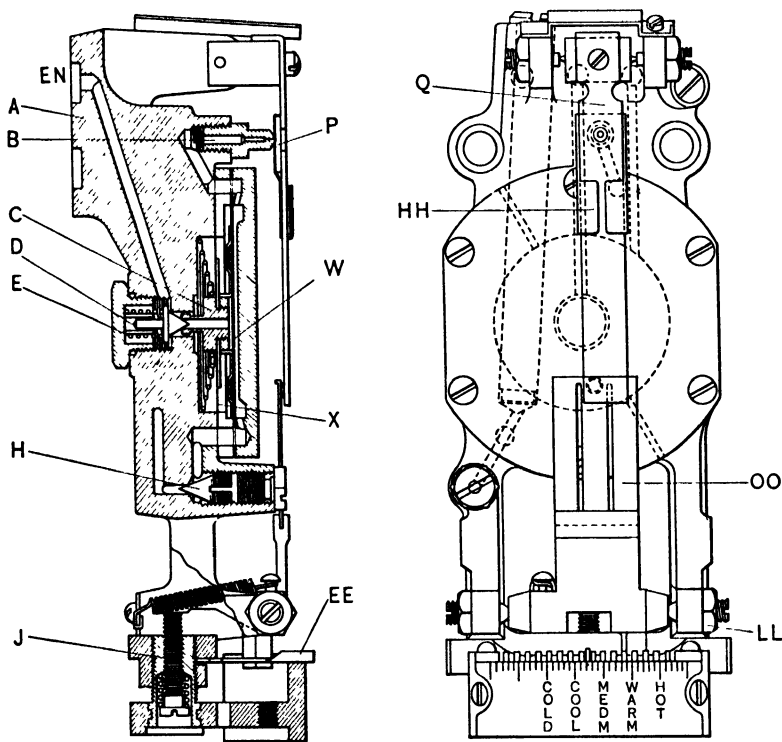
Courtesy of Johnson Service Co.

FIG. 231.—Johnson Intermediate (Standard) Room Thermostat.

Should the room cool, the thermostatic plate *OO*, made up of two strips of dissimilar metals, will bend to the right and open up the leak port *B* by swinging the seat away from its outlet. If the leak port remains closed, pressure on the right of the diaphragm will gradually build up and move the diaphragm to the left, forcing the valve *D* from its seat, and air will enter the space to left of the second diaphragm *X*, from which it passes to a passage delivering the air to the pipe system, leading it to the Johnson syphon valves or other air motors. This air acting on the bellows closes all the radiator valves on the line. When the room cools, the thermostatic plate bends to the right and moves the seat to the

right, uncovering the leak port. The leakage of this air from the right of the diaphragm allows the main valve *D* to move to the right and close, but at the same time this motion opens up the passage in the exhaust seat *C*, so that the air to the left of *X* and that from the radiator pipes can escape to the space between the diaphragms *W* and *X*, from which it passes to the atmosphere through a small discharge passage.

In this operation there is differential action between the pressures on the opposite sides of the pairs of diaphragms so that as air is admitted



Courtesy of Johnson Service Co.

FIG. 232.—Johnson Intermediate (Supersensitive) Room Thermostat.

to the left it forces the diaphragms back to the right and throttles the air supply so that there may be a definite pressure in the line for a change of temperature of one degree in the space around the thermostat. This is called the sensitivity, and the standard is $7\frac{1}{2}\%$ change per degree difference in temperature, although other sensitivities can be obtained and some thermostats are made with adjustments for varying this quality.

The adjusting screw *J* for the bimetal strip and its movable nut can

be used to vary the temperature at which the leak port is uncovered, as this fixes the temperature at which steam must be admitted. This may be operated by the occupant of the room, or in some thermostats the temperature adjustment is made by a key. Restricted adjustment may be had on some of them. The pointer *EE* indicates the temperature setting. The cover of one type thermostat is shown in Fig. 231.

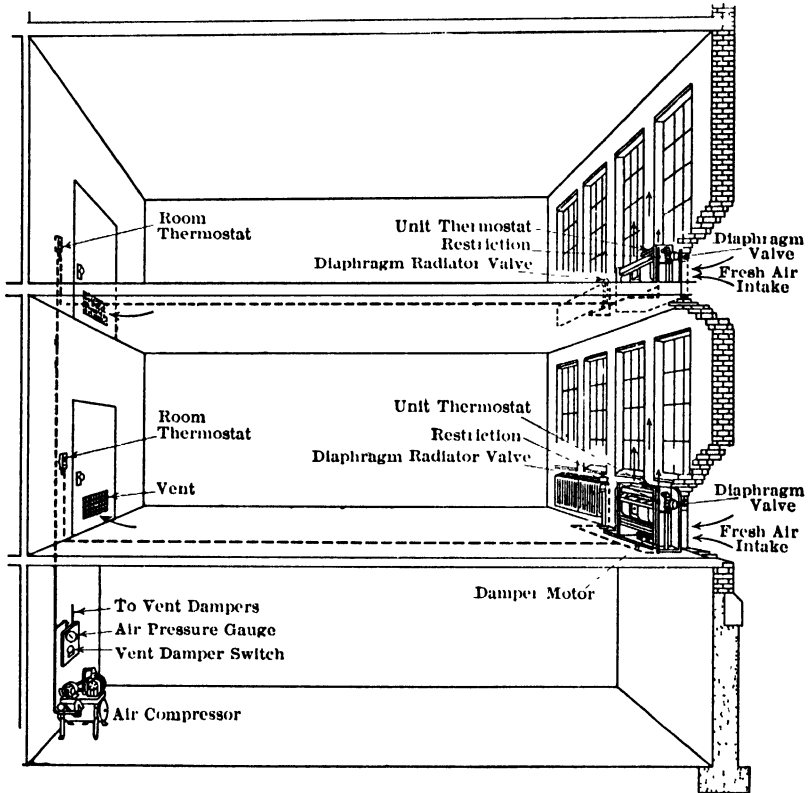
The Johnson Company builds thermostats which have a quick or positive action in place of the intermediate or graduated action just described, as well as a compound thermostat which combines both features. The dual thermostat has two expansion elements and seats, one set for day temperatures and one for night. The thermostats of a complete building may be set on one or the other control by the pressure in the pipe line acting on a separate bellows, and each thermostat may be manually changed from night setting to day setting if desired. When such thermostats must be set individually they are known as high-low thermostats. Another high and low thermostat is used for winter heating and summer cooling. An insertion thermostat is one in which the expanding element is placed in the medium for which the temperature is to be controlled while the compressed air element is placed outside. This medium may be air in a ventilating system or a fluid in a heating system.

For the regulation of humidity, the Johnson humidostats may be used to control the addition of steam to the ventilating air from a steam pipe humidifier, or from a pan in which water is boiled. The humidostat may vary the water temperature in an air washer by regulating the supply of medium to its heating or cooling coil. The devices are made in several forms. In one the action is quite similar to that of the thermostat, as a bar of two dissimilar woods or a membrane acts to close the leak port, on a change in relative humidity. In another form, the expansion member of an insertion thermostat is covered with a wet fabric and responds to the wet-bulb temperature of the flowing air.

The system of the Powers Regulator Company is another which has been used for many years. The application of this system to unit heaters combined with direct radiators is shown in Fig. 233. In this the room thermostats control the unit heaters and the direct radiators, although other thermostats are placed on the unit heater to regulate air temperature. The outlet vents shown at the floor line on the walls opposite the unit heaters are controlled by a compressed-air line operated by an air switch from a central point. The air compressor, storage tank and vent damper control line are indicated.

The Powers thermostat is shown in Fig. 234. The thermostatic element is a disc-like vessel with corrugated faces containing a volatile

fluid which responds to temperature changes, having a boiling point at atmospheric pressure of about 50° F and a 6 # gage pressure at 70° F. The pressure expands the disc and moves a pair of air valves. Compressed air enters a passage leading to a space around the supply valve. When the temperature rises the vapor disc expands and forces the exhaust valve assembly to the right against spring pressure, lifting the

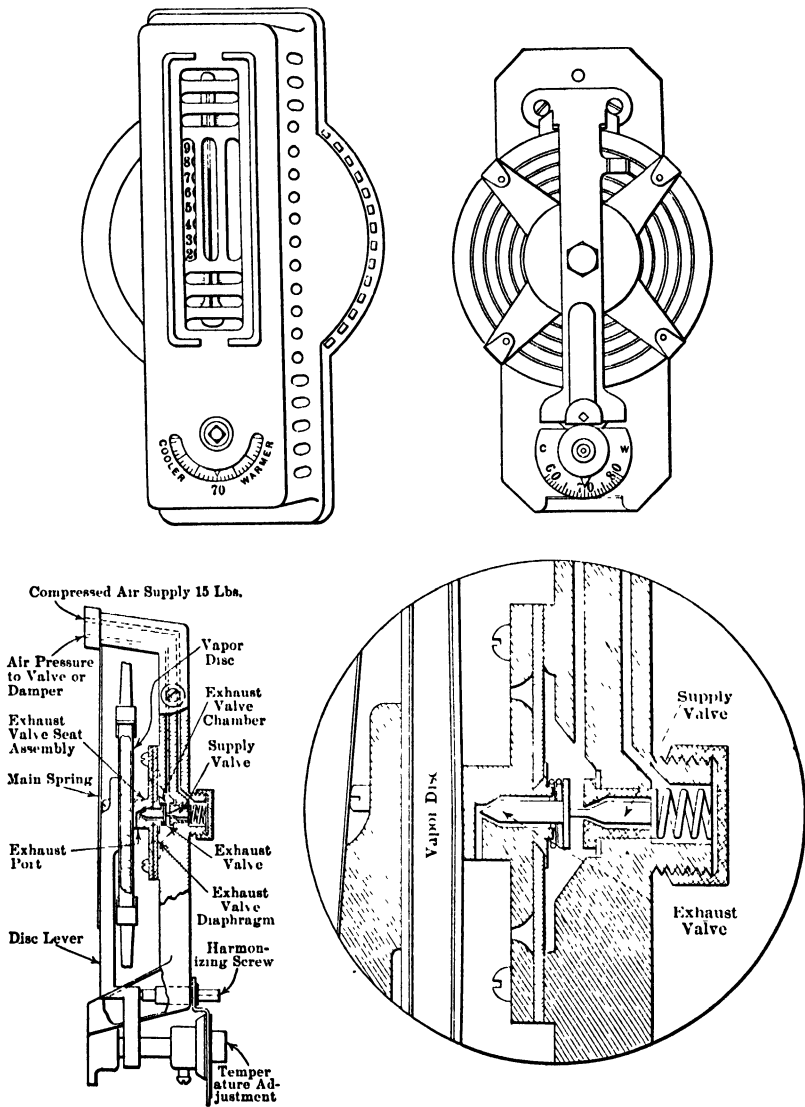


Courtesy of Powers Regulator Co.

FIG. 233.—Powers System of Temperature Control.

supply valve from its seat and allowing air to flow into the operating pipe line and the exhaust chamber from which it cannot escape into the exhaust port as the exhaust valve is closed. This air, exerting pressure on a diaphragm supporting the exhaust valve assembly, tends to return the supply valve to its seat and actually will do this when the pressure in this space bears a certain relation to that fixed by the boiling pressure of the volatile fluid at room temperature. The air which then flows from the space through a passage to the air pipes leading to the radiator

valves or other motors is fixed by the particular temperature at the thermostat, and this air closes the valves a definite amount depending



Courtesy of Powers Regulator Co.

FIG. 234.—Powers Thermostat.

on its pressure and the force of the valve spring. As the room cools, the disc will collapse, the supply valve will close and the pressure on the

exhaust valve diaphragm moves it to the left and the exhaust valve spring causes the valve to open and permit the discharge of air from the operating pipe system through the outlet passage. The steam valves will then open or the motors will operate by their weights or springs to move devices to the warming positions. The adjusting screw at the bottom moves the vapor disc support so that the temperature at which operations will take place may be varied.

The Powers thermostats are built for positive, gradual or compound action, for winter and summer operation and for day and night control. For duct work an extended disc or remote bulb may be used for temperature control.

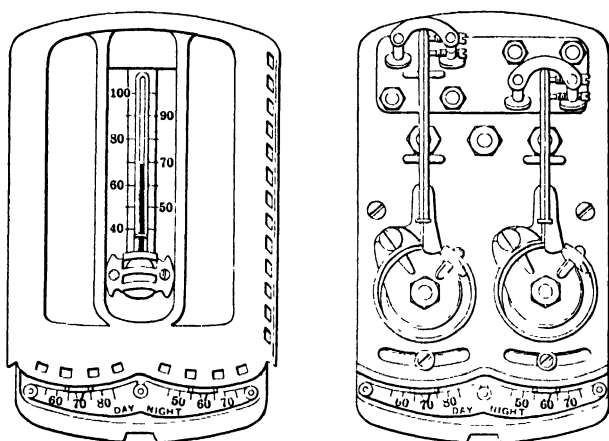


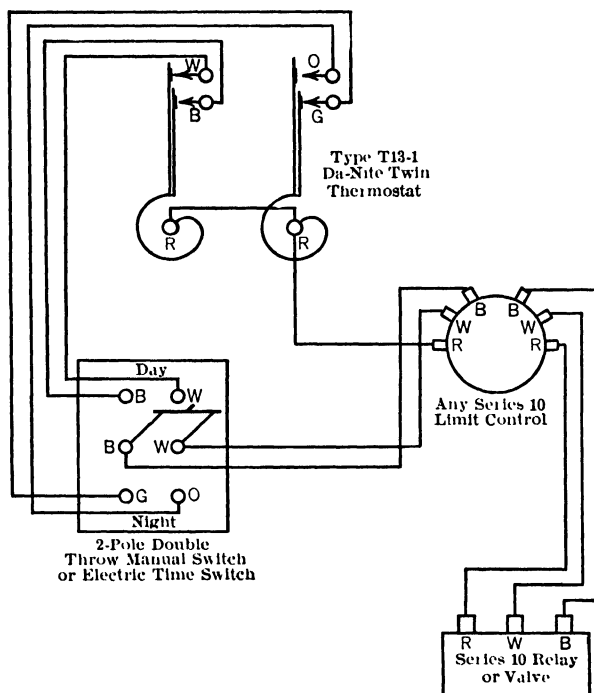
FIG. 235a.—Da-Nite Thermostat.

These companies build devices which vary the setting of thermostats or hygrostats of one part of a system with conditions at some other part, also devices which change settings or other conditions by clocks for a definite time program of operation and still others in which air switches are used to supply lines for multiple or individual operation of air motors.

In all of them the ability to regulate the air pressure discharged into the motor from the thermostat by the amount of temperature variation permits the holding of the motor in intermediate positions so that regulation may be accomplished by sustained action in place of frequent changes from a full open to a full closed condition. It also permits the engineer to operate devices in sequence by variation in the spring force on different units, although the use of relays or cumulators may make the sequence more definite.

Another system for the control of air conditions which has been in use for many years is that of the Minneapolis-Honeywell Regulator

Company. In the thermostats of this system the expansions and contractions of a bimetallic blade complete and break electric circuits which operate motors or solenoids, and thus control the flow of a thermal medium for heating, cooling or humidifying. The "da-nite" twin thermostat and its electric circuit are shown in Figs. 235a and b. The control switch at some chosen point is used to change from the day to the night setting and is operated by hand or by a clock.



Courtesy of Minneapolis Honeywell Regulator Co.

FIG. 235b.—Da-Nite Thermostat.

The pointer at the bottom indicates the operating temperature at which the thermostat is set as this may be varied by shifting the support of the blade.

A simpler single-blade thermostat is shown in Fig. 236 in which the white or blue contact with the blade circuit operates a motor to open or close the regulating device. The upper part of the thermostat receives a clock which may be arranged to shift it from the day setting to the night setting at any desired time, and these settings are adjusted by side pointers which operate on the contact base. The indicator at the bottom is used for setting the blade by the adjusting screw so that the

scale reading of the indicator is one degree lower than the reading of the thermometer on the case of the thermostat. There are two contacts on

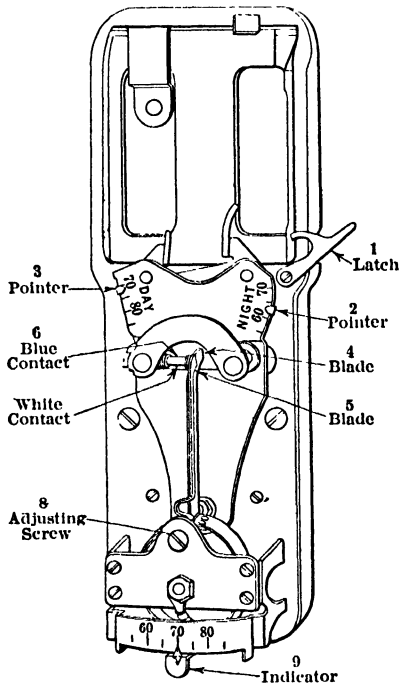
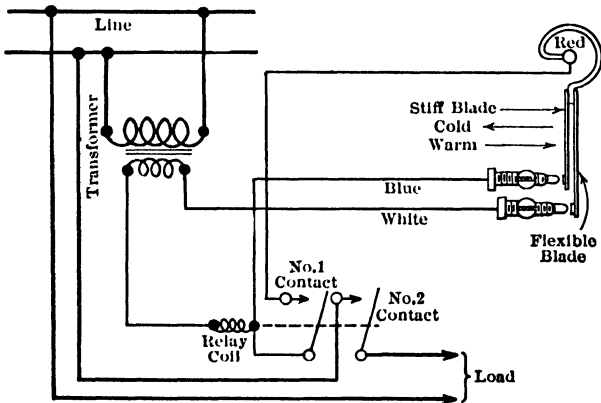


FIG. 236.—Electric Clock Thermostat.



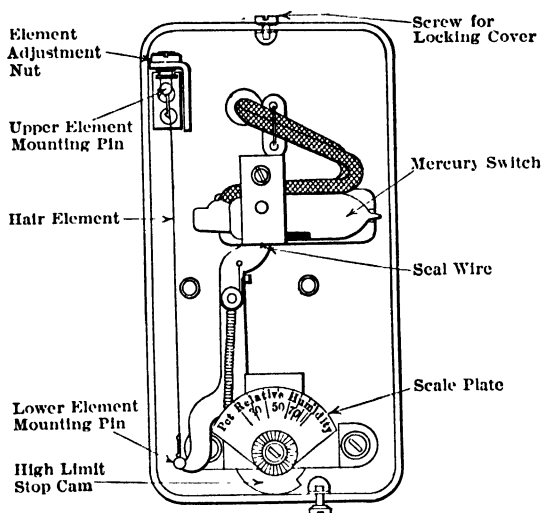
Courtesy of Minneapolis Honeywell Regulator Co.

FIG. 237.—Circuit for Electric Clock Thermostat.

each blade. The function of this double contact is to give a temperature difference between the on and off positions, shown by Fig. 237. The

orange, green, white and blue covered wires aid in arranging the circuits. The relay coil of the lower section which operates contacts 1 and 2 will close these after contacts are made by the blade on the white and then the blue point and the relay will hold the switch on after the blue point breaks, finally opening the contacts when the white contact is broken. The difference in temperature required to produce contacts at the white point and blue point may be adjusted to permit the desired temperature change and its time lag.

For humidity control an element of human hair operates a lever system containing the mercury switch, Fig. 238, as the relative humidity



Courtesy of Minneapolis Honeywell Regulator Co.

FIG. 238.—Humidity Control.

changes, and by adjusting the high limit stop cam the point at which the circuit to the humidifier valve is completed may be fixed to maintain a desired humidity.

The company manufactures controls for pressure as well as temperature, and one of their later developments is a weatherstat which is placed outside a building where it can respond to temperature, wind and sunshine. This consists of a covered mass of iron which is maintained at a fixed temperature by an electric warming element; the relation between the heat for this element and that for the building is so adjusted that these are proportional to each other, and whenever heat is required by the weatherstat heat is supplied to the building.

The motors of the Minneapolis-Honeywell Company, which drive

their motorized valves and dampers, are arranged to turn a crank arm through 180° , the circuits being such that, after the crank has turned through this angle, the operating circuit is broken but part of a new circuit is set for completion by contact at the opposite point of the thermostat, causing the shaft to be turned the remaining 180° at the proper time. Other motorized valves of the company are operated by solenoids.

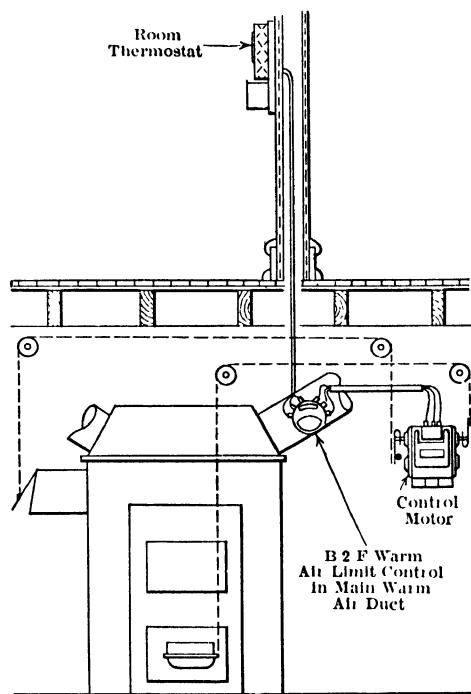


FIG. 239.—Minneapolis-Honeywell Regulator for Warm Air Furnace.

Fig. 239 illustrates the application of this system and its motor to a warm air furnace. In this case the room thermostat operates the check damper on the smoke flue on the left and the draft door of the ash pit whenever the warm air limit control permits this action, as this control closes the draft door and opens the check whenever the bonnet air is too hot.

In the air washer and heater, Fig. 240, the intake temperature control opens or closes the motorized steam valve of the preheater and the wet-bulb controller maintains the temperature of the washing spray water by acting on the three-way valve which is regulated by the humidity control or humidostat in the discharge flue. The temperature

in this flue is held constant by the duct thermostat which operates on the "modutrol" steam valve and the motor on the dampers associated with the reheater.

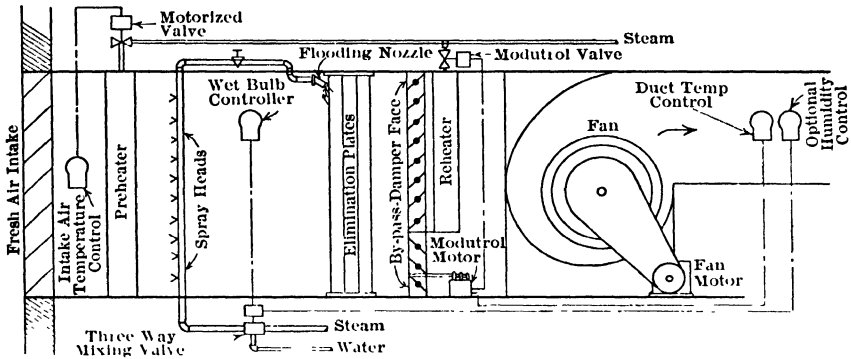
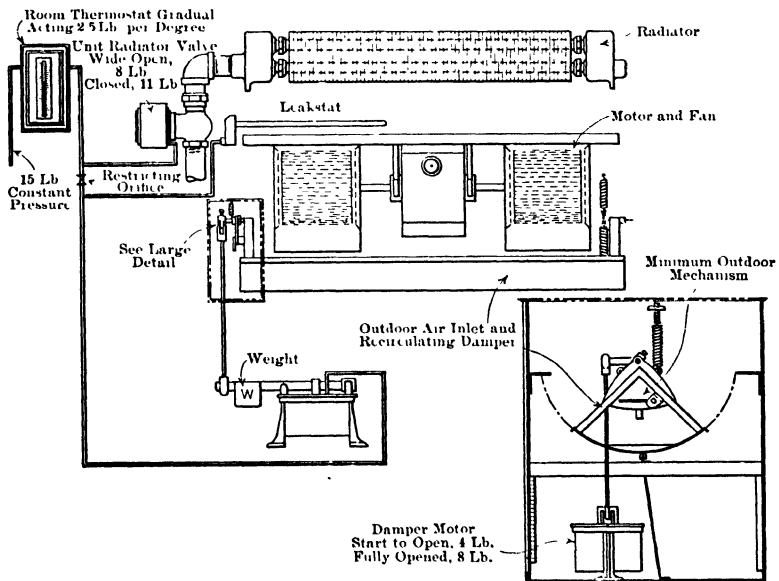


FIG. 240.—Minneapolis-Honeywell Regulator for Air Conditioner.



Courtesy of John J. Nesbitt, Inc.

FIG. 241.—Controls of Syncretizer.

The pneumatic control of the type O syncretizer of Nesbitt is shown in Fig. 241. In this unit heater the damper motor starts to operate on a pressure of 4% and completes its full stroke at 8% while the radiator

motor valve is equipped with a spring which holds the valve wide open until the pressure is 8% and at 11% this valve is completely closed. The leakstat shown in the figure is a device consisting of a tube of metal within which is a bar of another metal connected to an air valve. The relative expansions of these are such that whenever the temperature of the air surrounding it is below 60° the valve opens and allows air to discharge from the leakstat. A restricted orifice reduces the flow of air into the part of the air system containing the leakstat and leading to the damper motor.

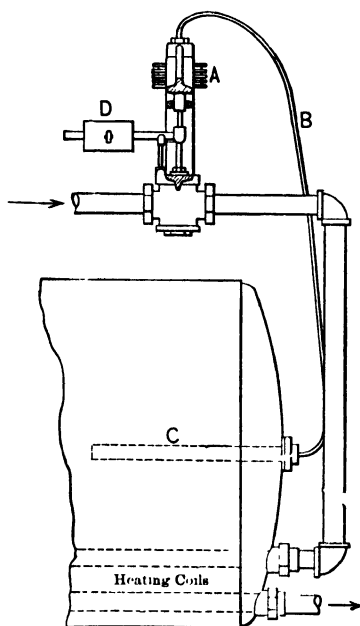


FIG. 242.—Volatile Liquid Thermostat.

With a room temperature below 67° F there is no air in the control piping beyond the thermostat, the steam valve is wide open and the weight on the damper motor has closed the damper to fresh air, so that on starting the fan motor air is recirculated from the room and the temperature of the room rises. When 67° F is reached by the air around the thermostat, the leak port is closed and air enters the system slowly, passes the restricted orifice and, as the leakstat is now closed, it starts to move the damper motor when 4% is reached in that section, the steam valve motor being unaffected. Fresh air enters from the outside, and if this is sufficient to reduce the temperature around the

leakstat to 60° F this device discharges air from the motor pipe system and the damper closes slightly from the weight. If the room temperature continues to rise, the sensitivity of the thermostat is such that at 69° F the pressure established by it has reached 8% and the radiator valve starts to close, reaching its fully closed position at 70° F.

Nesbitt also uses electrical controls such as those described under the Minneapolis-Honeywell system.

For controlling steam supply to coils of a water heater an insertion thermostat of one of the systems above or a volatile liquid thermostat, Fig. 242, may be used. The pressure of the vapor in the tube *C* which is fixed by the temperature must be sufficient to lift the weight *D* on an arm before the steam valve closes. The tube *B* and syphon bellows *A* form a closed system with the tube *C*.

PROBLEMS

1. A radiator of 70 sq ft of E.D.R. is placed at an equivalent length of 250 ft from the boiler in which the pressure is 1# gage. The system is designed for 0.25% per 100 ft. The return line at the boiler, 125 ft from the radiator, has a vacuum of 5 in. of mercury. Find the diameter of the orifice of this radiator. How much pressure should be carried at the boiler, with the same return vacuum carried, when only one-half of the maximum heat demand is required on the radiator and the other parts of the system? Pressure drop varies as square of steam flow.

2. The degree days from 65° F are given on page 352. How much steam should be charged for the month of January if the steam used for this season was 7250 M lb? Compute the degree days for the following average outside temperatures

Jan....15° F	Apr....45° F	July....80° F	Oct....52° F
Feb....10° F	May....60° F	Aug....80° F	Nov....48° F
Mar....18° F	June....68° F	Sept....65° F	Dec....30° F

3. Make sketches showing the location of thermostats and motors for controlling the building of Chapter V for direct steam, for hot water and warm air.

4. Make sketches showing the location of thermostats and motors for the apparatus used in the plenum system applied to the school building of Chapter IX.

CHAPTER XIV

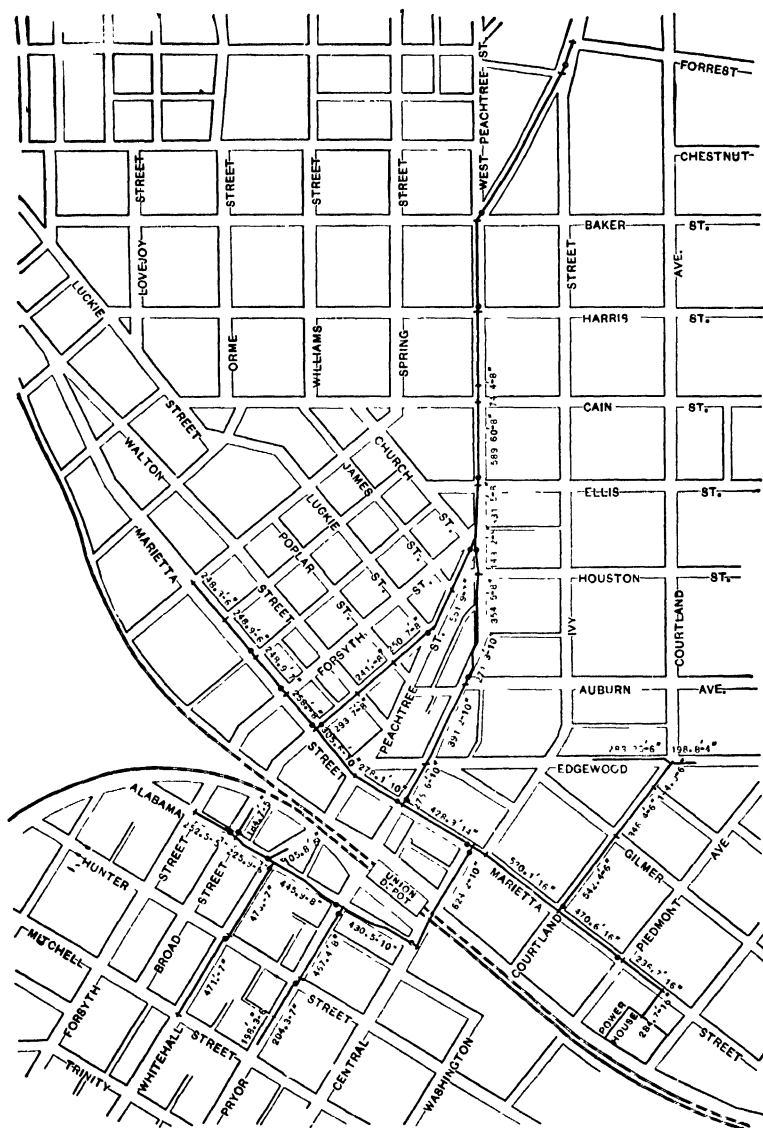
DISTRICT HEATING

District heating or heating from a central station has been extensively employed in various cities of the United States and in many institutions or manufacturing plants. The savings from a single efficient steam-generating plant supplying a number of buildings or the ability to use waste steam have combined with the elimination of dirt and troubles from ash removals to more than offset the yearly cost of the steam distribution system and its losses. The district heating in a city is shown in Fig. 243, in which steam from the power house on the railroad is taken through a 16 in. main and distributed over a large area, the most remote end of which is more than a mile from the power house. The steam lines reduce in size as the outer points are reached; 5 in., 7 in. and 8 in. are found at these places. Fig. 244 shows the district system of an educational institution in which three branches extend from the boiler house, some of the mains being carried in tunnels while others are carried through conduits.

When steam is used as the heating medium, it is carried at various pressures depending on the conditions of the system, realizing that, as the pressure is increased, its greater density makes it possible to carry a much greater weight of steam with the same pressure drop in a given pipe. The New York Steam Company delivers steam beneath the streets of New York at pressures of 150% so that it may be used for power purposes as well as for heating. This higher pressure and higher saturation temperature necessitate thick heat insulation on such lines. It should be realized, however, that even here, with the same insulation thickness, the larger pipe with low pressure and corresponding temperature has such a large surface that the heat loss from it is greater than that from a smaller pipe carrying the same amount of steam at higher pressure and temperature.

Exhaust steam from engines or other apparatus is sometimes available, and would be wasted were it not used in the building containing the apparatus or in the district heating system connected therewith. In such a case the low-pressure steam must be carried through large steam mains, as with low pressure it must be carried with a very small pressure drop.

District heating may be accomplished by hot water. Although the heat capacity of a pound of water is much less than that of a pound of



Courtesy of American District Steam Co.

FIG. 243.—District Heating for a City.

steam, the density is so much greater that less volume is needed for a given quantity of heat and smaller lines are required for such capacities.

a piston pump is used but with a centrifugal pump there will be a limited rise in pressure as a quantity of water is reduced. If the system remains completely closed at all points it will be remembered that the pressure for the height of water in the supply risers will be balanced or exceeded by that in the cold water return risers, although boosters may be used in a building of appreciable height. The water in such systems may be heated by fuels in water heaters of sectional or tube form or by coils

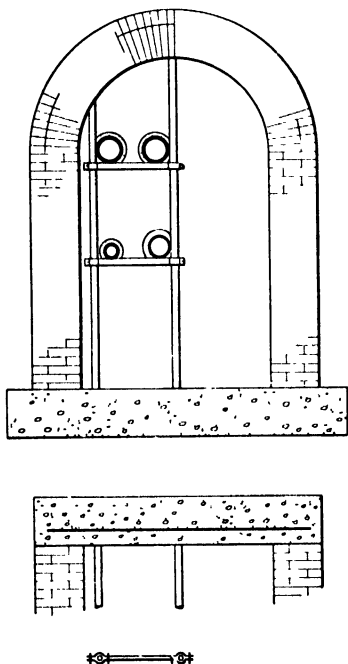
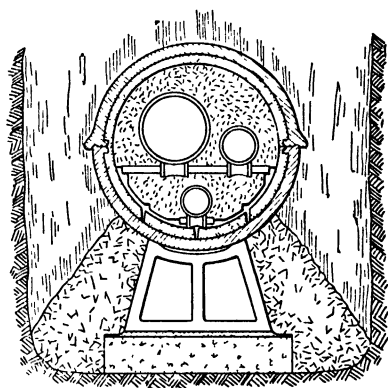


FIG. 245.—Pipe Tunnel and Supports.



Courtesy of Ric-wil Co.

FIG. 246.—Special Split Tile Pipe Conduit.

or tube heaters with exhaust or live steam. If this water is heated from fuel a thermostat will regulate the amount of combustion to maintain the desired outlet temperature. If exhaust steam is used the heater should be employed to circulate the water from the storage tank so that all exhaust steam is utilized.

In the steam system of district heating, the medium is delivered to the mains at the designed pressure, and at each point of heat utilization a branch delivers the supply for that center. The condensate from this center is returned through a return main if conditions permit. If there is a drop in elevation from all centers to the power plant, the condensate

may be returned by gravity; otherwise, the condensate from each center or that from a number of centers (if these are connected to a common return tank) may be pumped back to the power house. This return condensate has a heat value for boiler feed if its temperature is above that of the normal water supply for the power house. Whether or not this condensate return should be installed is a matter of cost. The yearly saving in the cost of water plus the value of the heat of the return above that for the temperature of the supply is compared with the yearly cost on the investment on this return line and its installation cost plus that of operation and pumping. If the latter is the greater, the return is delivered to the sewer. When the condensate from a building is delivered to the sewer it is taken through a coil for heating water or air before discharge so that it may be delivered to the sewer at as low a temperature as possible giving the maximum utilization of the heat supply.

Of course, the tunnel is advisable wherever the cost will permit its use as the pipes are always in position for inspection and repair and other facilities may be carried in the same structure. These tunnels should be made with sufficient space for maintenance, repair and alterations, and although a crowded tunnel has some advantages, its usefulness is not great when repairs must be made. The author recommends a clear space of at least 3 ft at the side and a minimum height of 6 ft. The floor of the tunnel should pitch to a gutter at one edge, and this should drain to sumps at which there are sewer connections or sump pumps to lift the water. The walls may be made of 12 in. brick work or of reinforced concrete to care for earth pressure on the side, the roof being made of a brick arch or a reinforced slab. At some point in the tunnel the roof should be designed with a proper removable slab of such length that sections of pipes may be introduced when repairs or extensions are required.

Pipes are supported on racks which are sometimes made of pipes and fittings. These are difficult to align and almost impossible to alter. A far better rack is one formed of pipe or I-beam stanchions at 16 ft intervals to which pairs of cross members are clamped by bolts as shown in the figure. These clamped pieces may be aligned easily and spaced to suit distribution of pipes, and after their final clamping they may be tacked to the uprights by welding. When a new spacing is desired the clamps may be cut and moved to a new position. The clamps should carry roller supports on which the pipe can move with little friction and the pipe is anchored at definite points, so that the expansion will occur between anchors at the expansion devices such as shown in Fig. 115.

The amount of expansion in 100 ft is given in Table 6-12, and from it the distance between anchors may be found from the amount of expansion permitted by the joint. In general the slip joints will permit movements of 7 in., variators 5 in., corrugations $\frac{1}{4}$ in. per convolution and pipe bends will expand distances given in the table below. The allowances in inches for each quarter bend of different pipe sizes are given with the radii of the bends to obtain them.

TABLE 14-1

EXPANSION CARED FOR BY QUARTER BENDS IN INCHES

Radius of Center Line of Bend, Inches	Pipe Sizes in Inches										
	1	2	3	4	5	6	7	8	10	16	20
12	0.25	0.12									
15	0.37	0.25	0.12								
20	0.75	0.50	0.37	0.25							
30	1.75	1.00	0.62	0.50	0.37	0.37					
40	3.12	1.75	1.12	1.00	0.75	0.62	0.50				
50	2.75	1.87	1.50	1.13	1.00	0.75	0.62			
60	3.87	2.62	2.00	1.62	1.37	1.00	0.87	0.75		
70	5.37	3.62	2.87	2.25	1.87	1.50	1.12	1.00		
80	4.75	3.75	3.00	2.50	1.87	1.50	1.37	0.87	
90	6.00	4.75	3.75	3.12	2.50	2.00	1.62	1.25	
100	5.75	4.62	3.87	3.00	2.37	2.00	1.50	
110	5.62	4.75	3.62	2.87	2.50	1.87	
120	5.62	4.37	3.50	2.87	2.25	1.75

Courtesy of Crane Co.

The pipes in a tunnel may be joined in any manner; screwed, flanged, or welded, as desired by the designer. Welding may be done very successfully and is convenient.

Anchors should be placed at points from which branch lines are taken, and if this is not done the branch must be arranged to allow movement in the line.

Having provided for expansion, the next important item of design is drainage. This is done by pitching the line $\frac{1}{16}$ in. to the foot in the direction of flow with drips at intervals of 500 ft. If the main of proper diameter is to pitch against the direction of flow, drip pots built up of a tee and a capped nipple 12 in. long, or such a nipple with a welded cap and welded to the pipe, should be placed at intervals of 200 ft.

The insulation of tunnel pipe is similar to that of any steam pipe as given in Chapter VI.

When pipes for district heating are carried in conduits the first step is to form the lower part of the conduit on the proper uniform grade with the necessary aligned pipe supports at intervals of 10 to 16 ft. In some cases a drain of unglazed tile is set in crushed stone beneath the conduit

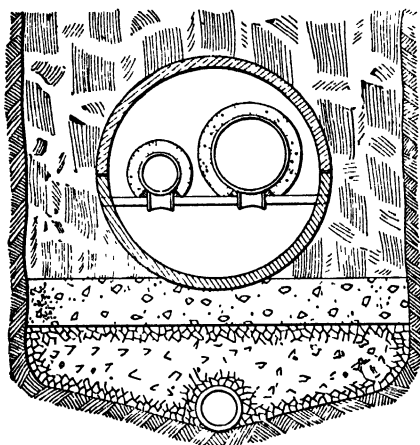


FIG. 247.—Split Tile Pipe Conduit.

with connections to a sump or sewer at intervals. A common form of conduit is made of split tile. These glazed terra-cotta pipes are molded and cut along a diameter, Fig. 247, before burning but are held together at the outer edge by the glazing. A knock from a hammer separates these halves, and after marking them to facilitate proper replacement of the top half, the bottom half is placed at correct elevation and earth is tamped along the sides to support this until the pipe is placed, tested and insulated. The pipe supports may be

formed in a special concrete half section or by cutting the lower half of the split tile for the reception of a pipe which forms the support for the pipe rollers. In each case the roller support must be put on the proper grade and blocked within the form into which the concrete is flowed.

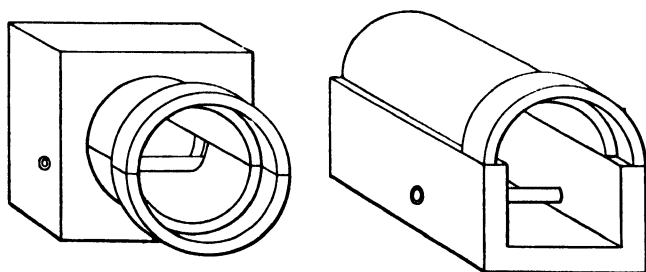


FIG. 248.—Pipe Supports in Conduits.

With split tile, the whole section may be placed in the trench without splitting after cutting the holes for the support. The section is set and aligned and then the support is aligned and fastened within the form for the mass of concrete which is now placed around the section. The final form of these support sections is shown in Fig. 248.

When the conduit is made of concrete the lower portion, Fig. 249, may be made with the sides of the trench acting as the outer forms. The inner form is made of wood, removed after setting for the reception of the pipe.

The pipe is now placed on the supports and the sections are welded or screwed together. Gasketed joints should not be used for pipe in conduits as the gaskets crack while in use and the leaks cannot be stopped without breaking into the conduit. After several hundred feet of pipe are laid the line should be tested with water pressure at $1\frac{1}{2}$ times the working pressure to make certain that joints are tight and that there are no holes in pipes or fittings. The pipe covering is now put on and carefully banded or preparation is made to fill the conduit with loose insulating material in bulk.

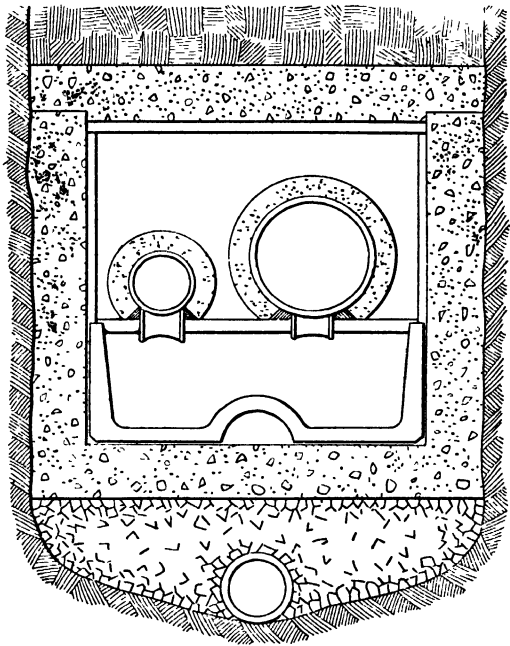


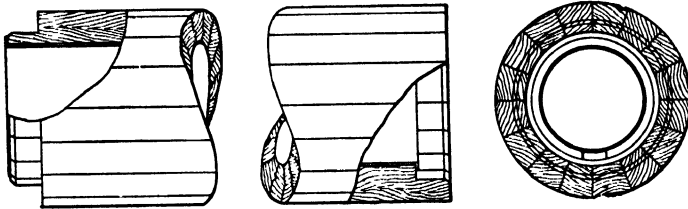
FIG. 249.—Concrete Pipe Conduit.

After this, the proper top half of the split tile is put on and the joints are made water tight with mortar of cement and sand or the top slab of concrete is cast over the sides, allowing the lower form for this to remain in the conduit. The earth is now back-filled and tamped with a flat rammer, and at times water is poured on the back fill to puddle it so as to compact this fill.

The American District Steam Company used a wooden pipe covering made of staves, Fig. 250, lined with tin and covered with a bituminous material. The sections of this covering fit each other with the equivalent of a bell and spigot on the ends which come together. The pipe is supported by rollers.

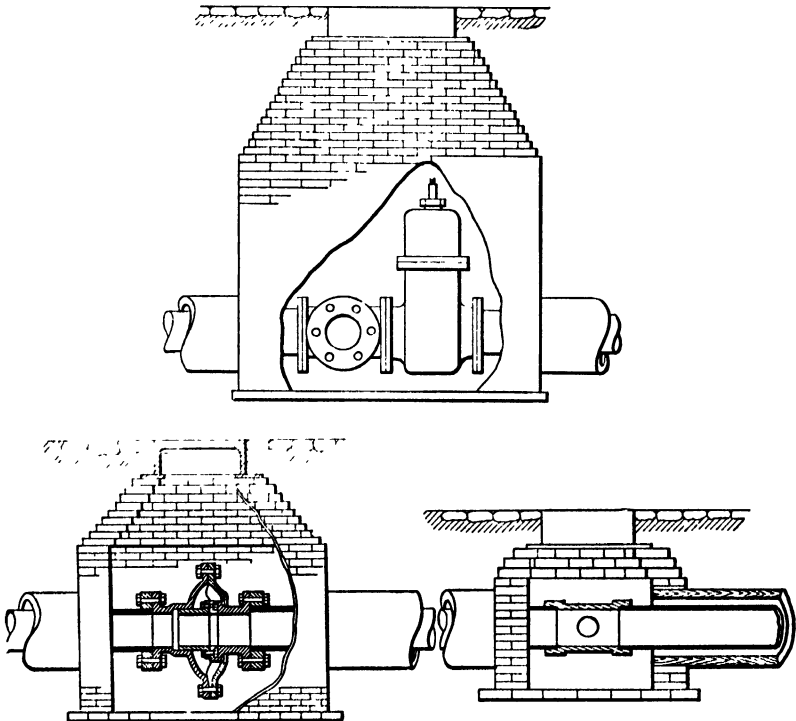
In conduit work, manholes are placed at points containing shut-off valves and branch outlets, and if these are at anchored points the branch may be taken directly from the tee with no swinging elbows for expan-

sion. Variators or slip expansion joints are placed in manholes, and as the bodies of these joints are fixed in position branches can be taken from them. The lower part of a variator is dripped to care for con-



Courtesy of American District Steam Co

FIG. 250.—Wooden Pipe Conduit.



Courtesy American District Steam Co.

FIG. 251.—Manholes for Conduit System.

densation. Fig. 251 illustrates the form of manholes used by the American District Steam Company. The variators are used at 50 ft intervals, if single; double ones care for lengths of 100 ft between anchors.

The manhole cover must always be circular in form.

Radiation loss from conduits is less than that from tunnel pipes as the earth covering prevents the loss of heat and the temperature around the pipe becomes so high that the heat loss is low. Experiments indicate that the heat loss per hour is given by

$$q = R \times \text{sq ft of external pipe surface} \quad (14-1)$$

$$q = \text{Btu per hr}$$

R is a constant depending on the temperature of the pipe, and given in Fig. 252 for different fluid temperatures.

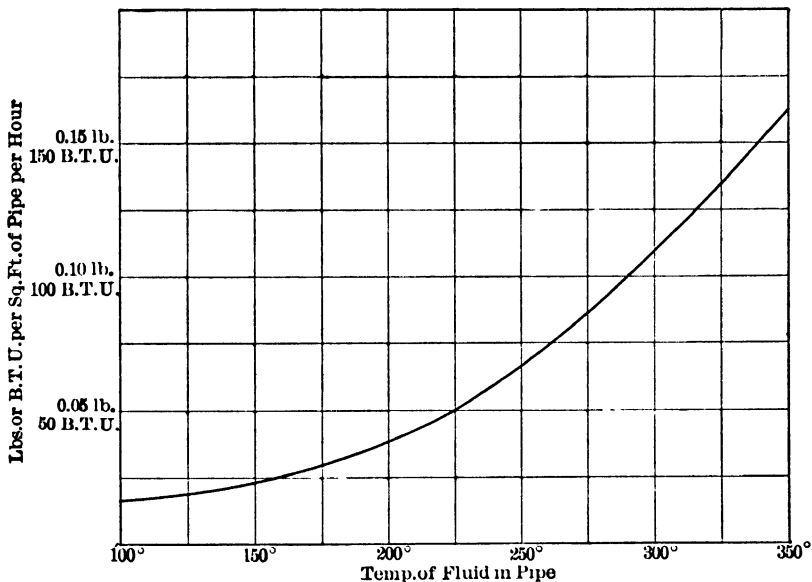


FIG. 252.—Coefficient for Conduit Heat Loss.

The charges made for heating from a commercial district heating system are based on the amount of steam used or on the amount of space to be heated or the square feet of radiation of the building. The first is called the meter method; the others, known as flat-rate methods, are not advisable for either consumer or company, as there is no incentive for the customer to use his service economically and as a result the company must make the flat rates high to compensate for the customer who wastes heat. With a metered service, however, payment is made for that which is used.

When the flat rate is used for a season, such as 45 cents per sq ft of steam radiation or 30 cents per sq ft of hot water radiation, an allowance

must be made if the consumer uses heat for a portion of the season which in some cases extends from October 1 to May 31. The company fixes these allowances, which should be proportional to the degree days of each month. At one time Gifford suggested the following percentages of charges for the different months:

TABLE 14-2

MONTHLY STEAM PERCENTAGES FOR HEATING

October	3%	February	19%
November	12%	March	13%
December	18%	April	8%
January	21%	May	3%

Total 97%

From degree days in one northern city the author has used the following:

October	5%	February	20%
November	10%	March	15%
December	15%	April	10%
January	20%	May	5%

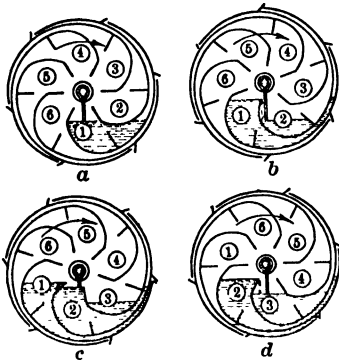
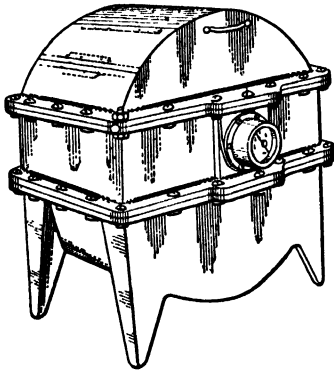
Of course, the better method of making the charge for heating is to base it on the amount of steam used. This amount is determined by metering the steam entering the building or by metering the condensate leaving it. The second method of metering has two drawbacks: the meters will not measure any leaks which occur in the building, and unless the meters are built to measure very low flows of water (condensate) some of the condensate will pass through the meter with no record.

One of the successful meters for measuring condensate is the Cadillac condensation meter, Fig. 253, in which the water from a constant flow trap, which enters at the center, falls from this pipe into a compartment and causes the drum made up of a number of these compartments to turn clockwise. By the time the falling water which was discharging into compartment 1 starts to drop into compartment 2, water from 1 begins to spill over into 2 and rises in the discharge neck leading from 1 to the periphery. When the level of the lip between 1 and 2 is such that it is at the elevation of the discharge lip from 1, no further water will discharge from 1 to 2 and the volume of the water in compartment 1 will now be completely discharged from the discharge lip as the drum rotates. Although the falling water disturbs the surface on which it falls, the final condition of the water surface is quiet when the discharge

position is reached, and the meter is an accurate volume measurer at a mere dribble or at the highest rate for which it can be used. The discharge being downward from the shaft center means that there is no turning torque from the impact of the water. The turning torque is derived from the weight of water in any compartment and the position of its center of gravity. To interpret this volume into weight the tem-

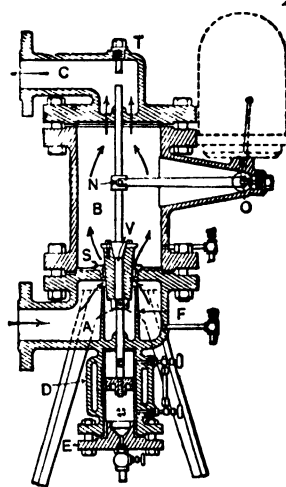
perature of the water must be known, and to be fair to the customer and heating company an endeavor should be made to cool this water by coils to some average temperature at which the weight calibration could be made.

When steam at high pressure is used in engines and pumps or even



Courtesy of Central Station Steam Co

FIG. 253.—Cadillac Condensation Meter.



From Steam Power Plant Engineering by Gebhardt

FIG. 254.—St. John Steam Meter.

for heating, leakage may occur, and then supply-steam meters are required. For measuring this steam a number of different meters are available and the various steam companies have selected the type which is accurate and reliable and for them appears desirable.

The St. John meter, Fig. 254, has been used by some companies. In this meter the amount of steam required at the outlet will be fixed by pressure drop between inlet A and outlet B and the area through which the steam flows. The weight of the conical plug valve V being fixed,

the pressure difference is fixed, and whenever the pressure in *B* tends to fall below this difference the conical plug is raised, increasing the area and permitting more steam to discharge. The drop in pressure being fixed, the quantity of steam of given pressure and quality depends on the area, which in turn depends upon the lift of the plug. This increase of area and weight can be made proportional to the movement by properly shaping the plug so that the movement of the pen above *O* driven from the plug will indicate the weight of steam in a desired unit of time. If this rate is recorded as the ordinate on a diagram on which time is the abscissa, due to the clock movement of the chart, the quantity of steam in a given time may be determined by the area under the record line. These meters may be so constructed that an integrating device will totalize the flow although the record sheet is needed to obtain the maximum rate of flow in any period.

In the layout of a district steam-heating system, the heat demand in Mbh of each building is determined, or if this is not known the square feet of radiation may be found. These are converted to pounds of steam per hour by using an average value of 240 Btu per hr or $\frac{1}{4}$ lb of steam per hr per sq ft of radiation. By starting at the most remote building the amount of steam flowing per hour or the square feet of E.D.R. at each center of heat demand can be found and tabulated together with the distances between these centers. With the pressure desired at the end or most remote point, the pressure drop between centers may be assumed, and from this, the steam flow and the density of the steam, the size of the pipe to carry the weight of steam between the sections may be determined by Eqs. (6-11), (6-3), (6-4) or (6-5). Table 6-5, for sizes with given heating surface, may also be used when the surface is adjusted for the pressure drops and densities which differ from those of the table. This is done for each section until the power house is reached, at which point the pressure to be carried will be known. Following this, other final pressures and pressure drops may be selected, giving a number of different pipe sizes for each of which the yearly cost of fixed charges, maintenance and operation must be computed, including insulation and heat loss. These final pressures at the boiler house may change the cost of the boiler, and this must be considered in the yearly cost. The solution giving the lowest yearly cost is the one to use.

An estimate may be made of probable heating surface for a projected building by taking 70% of its cubage from outside dimensions and then applying the heating surface ratios.

The size of the return piping carrying condensate is selected by a similar method, using the formulae of Chapter VII for friction losses and pumping pressures, obtaining the size which gives the smallest yearly

cost for fixed charges and costs for pumping and heat loss. If a gravity return is desired, the available head and quantity of return fix the size of the pipe.

If the steam pressure were fixed at the station by the present boiler equipment, or by the fact that exhaust steam from a turbine or engine would have to be used or that process steam were available, then the problems would be started at the power house, and by assuming the pressure needed at the end of the line, the drop per foot would be found and the pressure at each center of heat demand. With the pressure drop and length between centers, the density of the steam and the amount of steam carried in this section, the size of pipe could be found. In this case there can also be several solutions unless there is very little pressure difference between the power house and the end of the line.

For hot water district heating, the method of procedure would be the same, using the Mbh of demand at each center with the formulae of Chapter VII.

It should be the practice of the designer to consider the possible additions to the system, and many pipe lines are made larger than current demands because of this.

With the size of pipe fixed for tunnel or conduit, the heat loss from the mains can be found, and if this is added to the heat demands of the buildings or projected buildings, or the available buildings for a new central heating system, the total heat at the boiler house in Mbh is known.

Although the heat loss from the steam lines in a building is not always included in selecting the boiler or water heater for a building such as that of Chapter XII, the heat loss from the tunnel or conduit must be used in the selection of the boiler for a central system.

The total Mbh required for the district system and the data sheets of boiler manufacturers determine the sizes of the boiler or boilers for the power house, if heating boilers are used. If power boilers are to be employed it is customary to figure the boiler horsepower required. The total heat in Mbh may be changed to boiler horsepower by dividing by 33.472. Boilers are usually rated by allowing 10 sq ft of steam-making surface to a boiler horsepower, although in most plants the boiler is operated so that each nominal unit of capacity will produce several actual boiler horsepower in the steam. The ratio between the output in the boiler and its nominal output is expressed as per cent rating, so that for a given district system the nominal boiler horsepower needed will be

$$\text{Nominal boiler horsepower} = \frac{\text{Max Mbh} \times 100}{33.472 \times \text{max \% rating}} \quad (14-2)$$

Power boilers may be bought of various capacities from 75 boiler hp to 3000 or more bhp. In selecting the size of the boiler it is well to have at least two or three boilers carry the load, and then add a spare one, unless this gives units exceedingly large. With a large number of boilers it may be well to allow for the repair of one of them by increasing the rating of the others during repair. A rating of 200% to 300% may be used with good efficiency, and with four boilers in a plant, an increase from 200% to 270% would carry the load on three boilers. Boilers are often operated at 400 to 600% of nominal rating.

The power of boiler units having been fixed, the heating surface of the boiler is found by the formula:

$$\text{Square feet of heating surface} = 10 \times \text{Nominal boiler hp} \quad (14-3)$$

This number of square feet should be checked from the proposal of the boiler builder.

Yearly Cost. Often it is necessary to know the consumption of fuel and the amount of steam needed per heating season; the thousands of pounds of steam per season may be found by multiplying the Mbh for the designed condition of low outside temperatures of all buildings in the system by hours per day times the degree days in the heating season from some base (65° F or 66° F), dividing the product by the difference between the base temperature and the low temperature used in the design.

$$M \text{ Btu for season} = \frac{\text{Required Mbh} \times \text{hr}}{(66 - t_o)} \times \text{degree days} \quad (14-4)$$

$$M \text{ pounds of steam required} = \frac{M \text{ Btu}}{h - h'_o} \quad (14-5)$$

h = enthalpy of entering steam;

h'_o = enthalpy of condensate;

t_o = temperature of outside used in design;

hr = hours of steam supply per day.

$$\text{Pounds of fuel required} = \frac{M \text{ Btu for season} \times 1000}{\text{Heat per lb} \times \text{Eff.}} \quad (14-6)$$

The cost of the heat may be found if the cost of steam is known. This is usually computed by the operator of the plant and will vary from 30 cents per M lb to 70 cents per M lb at the power house, depending on the cost of fuel, labor, and fixed charges, as well as the conditions of the steam at delivery and the temperature of return condensate. When

this steam is delivered by a district heating company or by the plant itself, the cost of distribution should be added, and this, of course, includes the fixed charges and maintenance costs, on tunnels, conduits, lines and insulation. In many cases this brings the price to a much higher figure than the production figure given above. The cost of steam, as the cost of power, is determined by the cost of the fixed charges on the investment for the equipment to carry the peak load or maximum demand of the consumer plus the cost of labor, fuel, maintenance and supplies to produce the steam. The first element is called the demand cost, and the second the service or use cost. The former remains the same, regardless of the amount of steam consumed; the latter depends on the amount of steam used. In considering the demand cost the element of office expense for keeping accounts of the customers and of the plant should be included as this in general is independent of the amount of steam used.

Having established a demand cost and a service cost, the first may be expressed in dollars or cents per thousand pounds of steam per hour or day at the maximum demand during the month; the second, as the cost for the actual steam used at some rate per thousand pounds. These costs may also be covered by a minimum charge and a graduated cost per thousand pounds of steam, depending on the amount. Thus a service may be offered in which the following schedule would hold:

TABLE 14-3
RATE FOR COST OF STEAM

For the first	50,000 M lb per month	\$1.75 cents per M lb							
For the next	50,000 " " " "	1 30	"	"	"	"	"	"	"
For the next	100,000 " " " "	1 10	"	"	"	"	"	"	"
For the next	300,000 " " " "	0 95	"	"	"	"	"	"	"
For the next	500,000 " " " "	0 85	"	"	"	"	"	"	"
For the next	1,000,000 " " " "	0.75	"	"	"	"	"	"	"
For all over	2,000,000 " " " "	0 70	"	"	"	"	"	"	"

There will be a minimum charge to the consumer for 15,000 M lb of steam in any one month.

Most schedules are based on unit cost of fuel to the steam company and require an adjustment if the cost of fuel differs from the cost used in making the rate. Thus the cost of steam per thousand pounds may be changed 1 cent for a change in the cost of coal of 8 cents from a specified cost of \$3.75 per short ton.

In a private plant it is well to measure the steam made and used and determine the cost of making 1000 lb of steam, breaking down this cost into its elements: cost of fuel, cost of labor, cost of maintenance, cost of water, cost of fixed charges.

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To compare operations of plants, the amount of steam or Mbh per degree day (or per year) per 1000 cu ft of heated space or per sq ft of heating surface is found and this may be reduced to cost for financial reports although the reduction makes the figures less comparable because of the variations in unit cost of labor, supplies and fuel.

As a guide the following data have been collected and are added for comparison purposes.

Cost of steam for M lb for office buildings:

Total production 240,000 M per year

Labor.....	8.5 cents
Fuel.....	30.9
Maintenance.....	3.7
Make-up water.....	0.2
Power.....	0.7
Fixed charges.....	17.0
	<hr/>
Total.....	61.0

Fuel oil cost 3.00 cents per gal (8 lb)

Cost of steam per M lb for institutional buildings:

Total production 278,000 M per year.

Labor	6.7 cents
Fuel	27.4
Maintenance	0.3
Make-up water...	0.6
Power.....	1.9
Fixed charges.....	10.0
	<hr/>
Total.....	46.9 cents

Coal at \$5.25 per short ton.

Pounds of steam per degree day from 65° F per 1000 cu ft of heated space

For office buildings	1.00 lb
For apartment houses	1.75 lb
For department stores.....	0.60 lb
For institutional buildings.....	1.20 lb

Radiation in tunnel and conduit system of certain

district system..... 18% of heat carried

M lb steam per sq ft radiation per year

For institutional buildings...	0.90 M
For apartment houses.....	0.60 M
For hotels.....	0.55 M
For offices.....	0.40 M
For stores.....	0.40 M

In applying district heating to buildings or even for isolated unit heating it is well to study the effect of diminished night heating. By plotting the difference between outside and inside temperature, against time, the area below this curve will be proportional to the heat loss from the building. If now the heat be cut off at 5 P.M. or reduced in amount, the heat loss is greater than the heat supply and the temperature of the building will decrease because heat loss must be supplied by the heat of the parts of the building. This decrease of the temperature of the building means that the heat loss from the building will be less on account of the smaller temperature difference. Even though much of the night heat loss may come from the walls and parts of the building as well as its contents, which will require replacement by steam, yet because of the decrease of temperature difference between the temperatures on the two sides of walls and windows during these hours there is a definite saving. The difference between the temperature of the building during the shutdown and the outside temperature is plotted as before against time and the area below it shows the heat demand under this new method of operation. The heat supply after shut-off comes from the building materials and the contents of the building, and these require extra heat before occupancy in the morning to supply that removed during the shutdown period but the total heat demand from the steam, however, is proportional to the area under the curve for the whole day. The actual saving can be ascertained also by reading the steam or condensate meters when operating the building with a reduced period and without one. The degree day ratio would bring these readings to comparable bases. For buildings in which there are night occupancy and use, certain rooms must be heated; a thermostatic control system is then required as the steam mains will have to carry the necessary steam for any night user. By the use of the double setting thermostats, the whole building may be put under night-time control and the night worker will use the manual change over for the room he occupies. Of course, in a sectionalized system the various supply pipes are adjusted to give the proper pressure for the supply to the sections occupied at night.

PROBLEMS

1. A district system is used for an institution in which the boiler house is placed in the center of the group and the steam and return mains are carried in a tunnel connecting the building. The list of square feet of heating surfaces is given

Building	Surface	Distance
A	5 000	To B 200 ft + 4 elbows
B	3 000	To C 300 ft + 8 elbows
C	10 000	To boiler house 75 ft + 6 elbows
D	8 000	From boiler house 75 ft + 4 elbows
E	15 000	From D 200 ft + 4 elbows
F	1 000	From E 100 ft + 2 elbows

Find the sizes of main to use in each section of the tunnel for supply and return for steam at 5% gage, 50% gage and 200% gage with a drop of $\frac{1}{4}$ lb per 100 ft. Find the product of diameter and steam temperature for each of these results. What does this show?

2. In problem above $\frac{1}{4}$ lb per hr of steam per square foot of radiation is used for 600 hr during January and there is a heat loss in the tunnels of 10% of the heat used. Find the amount of steam per season, referring to Table 14-2. Find the total amount of steam used per year. Find the monthly costs, utilizing rates in Table 14-3. What would be conduit loss? Compute probable yearly steam demand from institutional requirements per sq ft.

3. How much expansion will take place in one of the lines of problem 1 if pressure is 50% gage and the temperature during original installation was 60° F? At what intervals should slip expansion joints be placed if the movement in each is 7 in.? How large would you make the radii of the compound pipe bend of C, Fig. 115, to care for 7 in. expansion in an 8 in. main?

APPENDIX

TABLE A-1

SYMBOLS, ABBREVIATIONS AND DIMENSIONS

<i>A</i>	Area, sq ft	<i>G</i>	Area of glass surface, sq ft
<i>a</i>	Area, sq in.	<i>g</i>	Acceleration of gravity, 32 2 ft/sec ²
<i>a</i>	Unit conductance of air space, Btu per hr per sq ft per deg F	<i>h</i>	Head of fluid, ft or in.
<i>a</i>	Constant in expression for <i>U</i>	<i>h_f</i>	Enthalpy of fluid, Btu per lb
<i>b</i>	Constant in expression for <i>U</i>	<i>h_{fg}</i>	Enthalpy of evaporation, Btu per lb
Bar	Barometric pressure, lb per sq in. or inches of mercury	<i>h_g</i>	Enthalpy of dry saturated steam, Btu per lb
<i>C</i>	Specific heat, Btu per lb per deg F	<i>h_i</i>	Enthalpy of inlet air, Btu per lb of dry air and associated moisture
<i>C</i>	Constant for anemometer	<i>h_w</i>	Pressure head in inches of water
<i>c</i>	Constant	<i>h_o</i>	Pressure head in oz
<i>C_a</i>	Unit conductance of structure, Btu per hr per sq ft per deg	<i>h_r</i>	Enthalpy of room air, Btu per lb of dry air and associated moisture
cfh	Cu ft per hr	<i>h_s</i>	Static head, ft lb per lb
cfm	Cu ft per min	<i>h_v</i>	Velocity head, ft-lb per lb
cu ft	Cubic feet	hp	Horsepower, ft-lb per unit of time
cu in.	Cubic inch	hp-hr	Horsepower hour, ft-lb or Btu
<i>D</i>	Depth of air duct, ft or in.	in.	Inches
<i>D</i>	Density	ips	in. per sec.
<i>D</i>	Diameter of wheel	<i>J</i>	Joule's equivalent, 778 6 ft-lb
<i>d</i>	Diameter, ft or in.	<i>k</i>	Coefficient of conduction, Btu per lb per deg
<i>E</i>	Emissivity, ratio of emission of body to that of black body under same condition	<i>K</i>	Constant for air leakage
E.D.R.	Equivalent direct radiation	<i>K</i>	Constant for anemometer or fan
<i>e</i>	Base of Napierian system, 2.7183	<i>k_w</i>	Air space factor in foil channels
<i>e</i>	Elevation energy, ft-lb per lb	<i>k_h</i>	Height factor in foil channels
fpm	Ft per min	kw	Kilowatt
fps	Ft per sec	kw-hr	Kilowatt-hour
<i>f</i>	Unit conductance of air film, Btu per hr per sq ft per deg	<i>L</i>	Left
<i>f</i>	Friction flow factor	<i>L</i>	Length, ft or in.
<i>F</i>	Pressure loss due to friction in ft of substance or lb per sq ft or sq in.	<i>l</i>	Side dimension of duct, ft or in.
ft	Feet	lb	Pounds, mass
<i>G</i>	Mass velocity, lb per sec or hr per sq ft of flow area	M	One thousand
		Mbh	Thousand Btu per hr
		mi	Milli-inches

TABLE A-1

SYMBOLS, ABBREVIATIONS AND DIMENSIONS—*Continued*

mph	Miles per hour	t_r	Temperature of room, deg F
n	Exponent	U	Coefficient of heat transfer, Btu per sq ft per hr per deg
n	Number of sheets of foil	u	Intrinsic energy, ft-lb per lb
N	Rpm	u	Number of splitters
p	Pressure, lb per sq in., inches of fluid, oz per sq in.	V	Volume of room, cu ft
p_t	Saturation pressure of temperature t , lb per sq in.	V	Velocity, in fps, ips or fpm
q	Heat in Btu	V	Volume, cu ft
q_c	Heat conducted per hour, Btu per hr	v	Volume of 1 lb of material, cu ft per lb
q_r	Heat radiated per hr, Btu per hr	V_a	Volume of 1 lb of dry air, cu ft per lb
q_t	Heat transmitted per hr, Btu per hr	V_d	Volume of 1 lb of dry air, cu ft per lb
q_c	Heat per hr by convection, Btu per hr	V_s	Volume of 1 lb of dry air with its saturation moisture, cu ft per lb dry air
R	Gas constant, ft-lb per lb per deg	V_w	Velocity of wind in mph
R	Right	W	Weight as measure of mass, lb
R	Ratio of area enclosing radiator to actual surface	W	Wall area, sq ft
R	Unit resistance of partition	W	Width of duct in direction of radius of bend, ft
R	Radius of duct elbow	W	Mass per hr, lb per hr
r	radius to point	w	Mass of 1 cu ft, lb pr cu ft
RH	relative humidity	W_i	Weight of moisture to saturate 1 lb dry air
R.N.	Reynolds number	W_v	Actual weight of moisture associated with 1 lb of dry air, lb per lb dry air
rpm	Revolutions per minute	W'	Weight of water to saturate 1 lb of dry air at wet-bulb temperature
S	Entropy, Btu per lb per deg	x	Density relative to air at 70° F and 29.92 in.
sq ft	Square feet	x	Number of particular splitter
t	Temperature, deg F	$\#$	lb per sq in.
t	Temperature, dry-bulb, deg F	Δ	Symbol of difference
T	Absolute temperature, deg F	ρ	Relative humidity
t'	Temperature, wet-bulb, deg F	ρ	Radius of duct splitter, ft
t_c	Temperature, ceilings, deg F	ω	Solid angle
t_c	Temperature cold substance, deg F	μ	Absolute viscosity, lb per sec ft
t_h	Temperature hot substance, deg F	η	Efficiency
t_i	Temperature inside, deg F		
t_i	Temperature of recirculated air, deg F		
t_s	Mean temperature of walls, deg F		
t_o	Temperature outside, temperature outlet, deg F		

TABLE A-2

PROPERTIES OF AIR AND MOISTURE FOR SATURATION. BASED
ON GOODENOUGH'S PROPERTIES OF STEAM AND AMMONIA

Temperature	Pressure of Saturated Vapor		Weight of Saturated Vapor in Grains		Volume in Cubic Feet		Enthalpy per Pound Dry Air in Btu		Density Relative to Dry Air at 70° F		*Specific Heat of Mixture
	in. of Hg	lb per sq in.	per cu ft	per lb dry air	1 lb dry air	1 lb dry air with vapor	air alone from 0° F	air from 0° F moisture from 32° F	dry air	saturated air	Btu per lb
0	0.038	0.018	0.47	5.47	11.58	11.59	0.00	0.83	1.152	1.153	0.2413
5	0.049	0.024	0.61	7.10	11.70	11.72	1.20	2.28	1.140	1.140	0.2414
10	0.063	0.031	0.77	9.16	11.83	11.86	2.41	3.80	1.128	1.127	0.2415
15	0.081	0.040	0.98	11.77	11.97	12.00	3.61	5.33	1.116	1.115	0.2416
20	0.103	0.050	1.24	15.01	12.09	12.13	4.82	7.11	1.104	1.103	0.2418
25	0.130	0.064	1.55	19.08	12.21	12.27	6.03	8.94	1.092	1.091	0.2419
30	0.165	0.081	1.94	24.11	12.34	12.41	7.23	10.92	1.082	1.080	0.2420
32	0.181	0.089	2.12	26.47	12.39	12.47	7.72	11.78	1.077	1.075	0.2421
34	0.196	0.096	2.29	28.70	12.44	12.52	8.20	12.60	1.073	1.070	0.2422
36	0.212	0.104	2.47	31.09	12.49	12.58	8.68	13.43	1.069	1.065	0.2423
38	0.229	0.113	2.66	33.66	12.54	12.64	9.17	14.35	1.064	1.061	0.2425
40	0.248	0.122	2.87	36.41	12.59	12.70	9.65	15.15	1.060	1.056	0.2426
42	0.268	0.132	3.09	39.38	12.64	12.76	10.14	16.20	1.056	1.051	0.2427
44	0.289	0.142	3.32	42.55	12.69	12.82	10.62	17.18	1.052	1.046	0.2428
46	0.312	0.153	3.57	45.94	12.74	12.88	11.10	18.18	1.047	1.042	0.2429
48	0.336	0.165	3.83	49.58	12.79	12.94	11.58	19.23	1.043	1.038	0.2430
50	0.362	0.178	4.11	53.47	12.84	13.00	12.07	20.33	1.039	1.034	0.2431
51	0.376	0.185	4.26	55.52	12.87	13.03	12.31	20.89	1.037	1.032	0.2431
52	0.390	0.192	4.41	57.64	12.89	13.07	12.55	21.46	1.035	1.030	0.2432
53	0.405	0.199	4.57	59.83	12.92	13.10	12.79	22.04	1.033	1.028	0.2433
54	0.420	0.206	4.73	62.09	12.95	13.13	13.03	22.64	1.031	1.026	0.2434
55	0.436	0.214	4.89	64.43	12.97	13.16	13.28	23.25	1.029	1.023	0.2435
56	0.452	0.222	5.07	66.85	13.00	13.20	13.52	23.88	1.027	1.021	0.2435
57	0.468	0.230	5.24	69.35	13.02	13.23	13.76	24.50	1.025	1.019	0.2436
58	0.486	0.238	5.42	71.93	13.05	13.26	14.00	25.16	1.023	1.016	0.2437
59	0.503	0.247	5.61	74.60	13.07	13.30	14.24	25.81	1.021	1.014	0.2438
60	0.521	0.256	5.80	77.30	13.10	13.33	14.48	26.48	1.019	1.013	0.2439
61	0.540	0.265	6.00	80.2	13.12	13.36	14.72	27.18	1.017	1.011	0.2440
62	0.560	0.275	6.21	83.2	13.15	13.40	14.97	27.88	1.016	1.009	0.2441
63	0.580	0.285	6.42	86.2	13.17	13.43	15.21	28.60	1.014	1.007	0.2442
64	0.600	0.295	6.63	89.3	13.20	13.47	15.45	29.33	1.012	1.004	0.2443

* From Fan Engineering, Buffalo Forge Company.

TABLE A-2—Continued

PROPERTIES OF AIR AND MOISTURE FOR SATURATION. BASED
ON GOODENOUGH'S PROPERTIES OF STEAM AND AMMONIA

Temperature	Pressure of Saturated Vapor		Weight of Saturated Vapor in Grains		Volume in Cubic Feet		Enthalpy per Pound Dry Air in Btu		Density Relative to Dry Air at 70° F		*Specific Heat of Mixture
	in. of Hg	lb per sq in.	per cu ft	per lb dry air	1 lb dry air	1 lb dry air with vapor	air alone from 0° F	air from 0° F moisture from 32° F	dry air	saturated air	
65	0.622	0.305	6.86	92.6	13.22	13.50	15.69	30.09	1.010	1.002	0.2444
66	0.644	0.316	7.08	95.9	13.25	13.54	15.93	30.85	1.008	0.999	0.2445
67	0.666	0.327	7.32	99.4	13.27	13.58	16.18	31.65	1.006	0.997	0.2446
68	0.690	0.339	7.56	103.0	13.30	13.61	16.42	32.45	1.004	0.995	0.2448
69	0.714	0.351	7.81	106.6	13.32	13.65	16.66	33.27	1.002	0.992	0.2449
70	0.739	0.363	8.07	110.5	13.35	13.69	16.90	34.19	1.000	0.990	0.2450
71	0.764	0.375	8.33	114.4	13.38	13.73	17.14	34.97	0.998	0.987	0.2451
72	0.791	0.388	8.60	118.4	13.40	13.76	17.38	35.85	0.996	0.985	0.2453
73	0.818	0.402	8.88	122.6	13.43	13.80	17.63	36.75	0.994	0.983	0.2454
74	0.846	0.415	9.17	126.9	13.45	13.84	17.87	37.68	0.992	0.982	0.2456
75	0.874	0.430	9.46	131.4	13.48	13.88	18.11	38.61	0.990	0.980	0.2457
76	0.904	0.444	9.76	135.9	13.50	13.92	18.35	39.58	0.988	0.978	0.2458
77	0.935	0.459	10.07	140.7	13.53	13.96	18.59	40.57	0.986	0.975	0.2460
78	0.966	0.474	10.39	145.6	13.55	14.00	18.84	41.60	0.984	0.972	0.2462
79	0.998	0.490	10.72	150.6	13.58	14.05	19.08	42.64	0.982	0.970	0.2463
80	1.031	0.507	11.06	155.8	13.60	14.09	19.32	43.70	0.980	0.968	0.2465
81	1.066	0.523	11.40	161.2	13.63	14.13	19.56	44.80	0.979	0.966	0.2466
82	1.101	0.541	11.76	166.7	13.65	14.17	19.80	45.90	0.977	0.964	0.2468
83	1.137	0.558	12.12	172.4	13.68	14.22	20.04	46.05	0.975	0.962	0.2470
84	1.174	0.577	12.50	178.3	13.70	14.26	20.29	48.23	0.973	0.960	0.2472
85	1.212	0.595	12.89	184.4	13.73	14.31	20.53	49.43	0.972	0.958	0.2474
86	1.251	0.615	13.28	190.6	13.75	14.35	20.77	50.67	0.970	0.956	0.2476
87	1.292	0.635	13.68	197.0	13.78	14.40	21.01	51.89	0.969	0.953	0.2478
88	1.334	0.655	14.10	203.7	13.80	14.45	21.25	53.23	0.967	0.951	0.2480
89	1.377	0.676	14.53	210.6	13.83	14.50	21.50	54.57	0.965	0.949	0.2482
90	1.421	0.698	14.96	217.6	13.86	14.55	21.74	55.93	0.964	0.947	0.2485
91	1.466	0.720	15.41	224.9	13.88	14.60	21.98	57.33	0.962	0.945	0.2487
92	1.512	0.743	15.87	232.4	13.91	14.65	22.22	58.76	0.960	0.942	0.2489
93	1.560	0.766	16.34	240.1	13.93	14.70	22.46	60.24	0.958	0.939	0.2492
94	1.609	0.790	16.82	247.1	13.96	14.75	22.71	61.75	0.956	0.937	0.2494

* From Fan Engineering, Buffalo Forge Company.

TABLE A-2—Continued

PROPERTIES OF AIR AND MOISTURE FOR SATURATION. BASED
ON GOODENOUGH'S PROPERTIES OF STEAM AND AMMONIA

Temperature	Pressure of Saturated Vapor		Weight of Saturated Vapor in Grains		Volume in Cubic Feet		Enthalpy per Pound Dry Air in Btu		Density Relative to Dry Air at 70° F		*Specific Heat of Mixture
	in. of Hg	lb per sq in.	per cu ft	per lb dry air	1 lb dry air	1 lb dry air with vapor	air alone from 0° F	air from 0° F moisture from 32° F	dry air	saturated air	Btu per lb
95	1.659	0.815	17.32	256.3	13.98	14.80	22.95	63.31	0.955	0.935	0.2496
96	1.710	0.840	17.82	264.8	14.01	14.86	23.19	64.90	0.953	0.933	0.2499
97	1.763	0.866	18.35	273.6	14.03	14.91	23.43	66.53	0.951	0.930	0.2501
98	1.818	0.893	18.88	282.5	14.06	14.97	23.67	68.20	0.950	0.927	0.2504
99	1.874	0.920	19.42	291.8	14.08	15.02	23.91	69.94	0.948	0.925	0.2506
100	1.931	0.949	19.98	301.3	14.11	15.08	24.16	71.70	0.946	0.923	0.2509
101	1.990	0.978	20.56	311.2	14.14	15.14	24.40	73.53	0.945	0.921	0.2512
102	2.051	1.007	21.15	321.4	14.16	15.20	24.64	75.39	0.943	0.919	0.2515
103	2.113	1.038	21.75	331.9	14.19	15.26	24.88	77.32	0.941	0.917	0.2518
104	2.176	1.069	22.36	342.7	14.21	15.33	25.13	79.31	0.940	0.914	0.2521
105	2.241	1.101	22.99	354	14.24	15.39	25.37	81.24	0.938	0.912	0.2525
106	2.308	1.134	23.64	365	14.26	15.46	25.61	83.39	0.937	0.910	0.2529
107	2.377	1.168	24.30	377	14.29	15.52	25.85	85.53	0.935	0.908	0.2531
108	2.447	1.202	24.98	389	14.31	15.59	26.09	87.72	0.933	0.905	0.2535
109	2.520	1.238	25.67	402	14.34	15.66	26.33	89.99	0.931	0.902	0.2539
110	2.594	1.274	26.38	415	14.36	15.73	26.58	92.32	0.930	0.900	0.2543
112	2.748	1.350	27.85	442	14.41	15.87	27.06	97.15	0.927	0.895	0.2551
114	2.909	1.429	29.39	471	14.46	16.02	27.55	102.3	0.923	0.890	0.2559
116	3.079	1.512	31.00	502	14.52	16.18	28.03	107.7	0.920	0.885	0.2568
118	3.257	1.600	32.68	534	14.57	16.35	28.51	113.4	0.917	0.880	0.2578
120	3.444	1.692	34.44	569	14.62	16.52	29.00	119.5	0.914	0.874	0.2587
125	3.952	1.941	39.19	667	14.75	16.99	30.21	136.5	0.906	0.861	0.2613
130	4.523	2.221	44.49	780	14.88	17.53	31.42	155.9	0.898	0.846	0.2644
135	5.163	2.536	50.38	913	15.00	18.13	32.63	178.7	0.891	0.832	0.2680
140	5.878	2.887	56.91	1072	15.13	18.84	33.85	205.7	0.883	0.818	0.2721
145	6.677	3.280	64.1	1260	15.26	19.64	35.06	237.3	0.876	0.802	0.2767
150	7.566	3.716	72.1	1485	15.39	20.60	36.27	275.1	0.869	0.786	0.2820
160	9.649	4.739	90.6	2091	15.64	23.09	38.69	376.1	0.855	0.751	0.2950
170	12.20	5.990	112.8	3026	15.90	26.84	41.12	531.3	0.842	0.712	0.3121
180	15.29	7.51	139.4	4604	16.16	33.04	43.55	791.8	0.828	0.670	0.3351

* From Fan Engineering, Buffalo Forge Company.

TABLE A-3-1
SATURATED STEAM: PRESSURE TABLE

The properties of saturated and superheated steam are from Keenan's Steam Tables, and are reproduced with permission of the author and the publisher.

Abs. Press.		Specific Volume			Enthalpy			Entropy			Abs. Press.
Lb./Sq. In.	Temp.	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Sat. Liquid	Evap.	Sat. Vapor	Lb./Sq. In.
p	t	v_f	v_{fg}	v_g	h_f	h_{fg}	h_g	s_f	s_{fg}	s_g	p
$\frac{1}{2}$ " Hg	58.83	0.01603	1256.9	1256.9	26.88	1058.8	1085.7	0.0533	2.0422	2.0955	$\frac{1}{2}$ " Hg
$\frac{3}{4}$ " Hg	70.44	0.01605	856.5	856.5	38.47	1052.5	1091.0	0.0754	1.9856	2.0609	$\frac{3}{4}$ " Hg
1" Hg	79.06	0.01607	652.7	652.7	47.06	1047.8	1094.9	0.0914	1.9451	2.0365	1" Hg
$1\frac{1}{4}$ " Hg	91.75	0.01610	445.3	445.3	59.72	1040.8	1100.6	0.1147	1.8877	2.0024	$1\frac{1}{4}$ " Hg
$1\frac{1}{2}$ " Hg	101.17	0.01613	339.5	339.5	69.10	1035.7	1104.8	0.1316	1.8468	1.9784	$1\frac{1}{2}$ " Hg
$1\frac{3}{4}$ " Hg	108.73	0.01616	275.2	275.2	76.63	1031.5	1108.1	0.1450	1.8148	1.9598	$1\frac{3}{4}$ " Hg
2" Hg	115.08	0.01618	231.8	231.8	82.96	1027.9	1110.8	0.1561	1.7885	1.9446	2" Hg
1.0	101.76	0.01614	333.8	333.9	69.69	1035.3	1105.0	0.1326	1.8442	1.9769	1.0
2.0	126.10	0.01623	173.94	173.96	93.97	1021.6	1115.6	0.1750	1.7442	1.9192	2.0
3.0	141.49	0.01630	118.84	118.86	109.33	1012.7	1122.0	0.2009	1.6847	1.8856	3.0
4.0	152.99	0.01636	90.72	90.74	120.83	1005.9	1126.8	0.2198	1.6420	1.8618	4.0
5.0	162.25	0.01641	73.59	73.61	130.10	1000.4	1130.6	0.2348	1.6088	1.8435	5.0
6.0	170.07	0.01645	62.03	62.05	137.92	995.8	1133.7	0.2473	1.5814	1.8287	6.0
7.0	176.85	0.01649	53.68	53.70	144.71	991.7	1136.4	0.2580	1.5582	1.8162	7.0
8.0	182.87	0.01652	47.38	47.39	150.75	988.1	1138.9	0.2674	1.5379	1.8053	8.0
9.0	188.28	0.01656	42.42	42.44	156.19	984.8	1141.0	0.2758	1.5200	1.7958	9.0
10.0	193.21	0.01658	38.44	38.45	161.13	981.8	1143.0	0.2834	1.5040	1.7874	10.0
11.0	197.75	0.01661	35.15	35.17	165.68	979.1	1144.8	0.2903	1.4894	1.7797	11.0
12.0	201.96	0.01664	32.40	32.42	169.91	976.5	1146.4	0.2968	1.4760	1.7727	12.0
13.0	205.88	0.01666	30.06	30.08	173.85	974.1	1147.9	0.3027	1.4636	1.7663	13.0
14.0	209.56	0.01669	28.05	28.06	177.55	971.8	1149.3	0.3082	1.4521	1.7604	14.0
14.696	212.00	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	14.696
16.0	216.32	0.01673	24.75	24.76	184.35	967.4	1151.8	0.3184	1.4312	1.7496	16.0
18.0	222.40	0.01678	22.16	22.18	190.48	963.5	1154.0	0.3274	1.4127	1.7402	18.0
20.0	227.96	0.01682	20.078	20.095	196.09	959.9	1156.0	0.3356	1.3960	1.7317	20.0
22.0	233.07	0.01685	18.363	18.380	201.25	956.6	1157.8	0.3431	1.3809	1.7240	22.0
24.0	237.82	0.01689	16.924	16.941	206.05	953.4	1159.5	0.3500	1.3670	1.7170	24.0
26.0	242.25	0.01692	15.701	15.718	210.54	950.4	1161.0	0.3564	1.3542	1.7106	26.0
28.0	246.41	0.01695	14.647	14.664	214.75	947.7	1162.4	0.3624	1.3422	1.7046	28.0
30.0	250.34	0.01698	13.728	13.745	218.73	945.0	1163.7	0.3680	1.3310	1.6990	30.0
32.0	254.05	0.01701	12.923	12.940	222.50	942.5	1165.0	0.3732	1.3206	1.6938	32.0
34.0	257.58	0.01704	12.209	12.226	226.09	940.0	1166.1	0.3783	1.3107	1.6890	34.0
36.0	260.94	0.01707	11.570	11.587	229.51	937.7	1167.2	0.3830	1.3014	1.6844	36.0
38.0	264.16	0.01710	10.998	11.015	232.79	935.5	1168.3	0.3876	1.2925	1.6800	38.0
40.0	267.24	0.01712	10.480	10.497	235.93	933.3	1169.2	0.3919	1.2840	1.6759	40.0
42.0	270.21	0.01715	10.010	10.027	238.95	931.2	1170.2	0.3961	1.2759	1.6720	42.0
44.0	273.06	0.01717	9.582	9.599	241.86	929.2	1171.1	0.4000	1.2682	1.6683	44.0
46.0	275.81	0.01719	9.189	9.207	244.67	927.2	1171.9	0.4039	1.2608	1.6647	46.0
48.0	278.45	0.01722	8.829	8.846	247.37	925.4	1172.7	0.4076	1.2537	1.6613	48.0
50.0	281.01	0.01724	8.496	8.514	249.98	923.5	1173.5	0.4111	1.2469	1.6580	50.0
52.0	283.49	0.01726	8.189	8.206	252.52	921.7	1174.3	0.4145	1.2404	1.6549	52.0
54.0	285.90	0.01728	7.902	7.919	254.99	920.0	1175.0	0.4178	1.2340	1.6518	54.0
56.0	288.23	0.01730	7.636	7.653	257.38	918.3	1175.7	0.4210	1.2279	1.6489	56.0
58.0	290.50	0.01732	7.388	7.405	259.71	916.6	1176.4	0.4241	1.2220	1.6461	58.0
60.0	292.71	0.01735	7.155	7.172	261.98	915.0	1177.0	0.4271	1.2162	1.6434	60.0
62.0	294.85	0.01737	6.937	6.955	264.18	913.4	1177.6	0.4300	1.2107	1.6407	62.0
64.0	296.94	0.01739	6.732	6.749	266.33	911.9	1178.2	0.4329	1.2053	1.6382	64.0
66.0	298.98	0.01741	6.539	6.556	268.43	910.4	1178.8	0.4356	1.2001	1.6357	66.0
68.0	300.98	0.01743	6.357	6.375	270.49	908.9	1179.4	0.4384	1.1950	1.6333	68.0
70.0	302.92	0.01744	6.186	6.203	272.49	907.4	1179.9	0.4410	1.1900	1.6310	70.0
72.0	304.82	0.01746	6.024	6.041	274.45	906.0	1180.5	0.4435	1.1852	1.6287	72.0
74.0	306.68	0.01748	5.870	5.887	276.37	904.6	1181.0	0.4460	1.1805	1.6265	74.0
76.0	308.50	0.01750	5.723	5.741	278.25	903.2	1181.5	0.4485	1.1759	1.6244	76.0
78.0	310.28	0.01752	5.584	5.602	280.09	901.9	1182.0	0.4509	1.1714	1.6223	78.0
80.0	312.03	0.01754	5.452	5.470	281.90	900.5	1182.4	0.4532	1.1670	1.6202	80.0
82.0	313.74	0.01756	5.325	5.343	283.67	899.2	1182.9	0.4555	1.1627	1.6182	82.0
84.0	315.42	0.01757	5.204	5.222	285.42	897.9	1183.4	0.4578	1.1586	1.6163	84.0
86.0	317.06	0.01759	5.089	5.107	287.13	896.7	1183.8	0.4599	1.1545	1.6144	86.0
88.0	318.68	0.01761	4.979	4.997	288.80	895.4	1184.2	0.4621	1.1505	1.6126	88.0
90.0	320.27	0.01763	4.874	4.892	290.45	894.2	1184.6	0.4642	1.1465	1.6107	90.0
92.0	321.83	0.01764	4.773	4.791	292.07	893.0	1185.0	0.4663	1.1427	1.6090	92.0
94.0	323.37	0.01766	4.676	4.694	293.67	891.8	1185.4	0.4683	1.1389	1.6072	94.0
96.0	324.88	0.01768	4.584	4.602	295.25	890.6	1185.8	0.4703	1.1352	1.6056	96.0
98.0	326.37	0.01769	4.494	4.512	296.80	889.4	1186.2	0.4723	1.1316	1.6038	98.0

TABLE A-3-1—Continued
PRESSURES

Abs. Press. Lb./Sq. In.	Temp. Deg. F.	Specific Volume			Enthalpy			Entropy			Abs. Press. Lb./Sq. In.
		Sat. Liquid v _f	Evap. v _{fg}	Sat. Vapor v _g	Sat. Liquid h _f	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _f	Evap. s _{fg}	Sat. Vapor s _g	
100.0	327.83	0.01771	4.408	4.426	298.33	888.2	1186.6	0.4742	1.1280	1.6022	100.0
102.0	329.27	0.01773	4.326	4.344	299.83	887.1	1186.9	0.4761	1.1245	1.6006	102.0
104.0	330.68	0.01774	4.247	4.265	301.30	886.0	1187.3	0.4779	1.1211	1.5990	104.0
106.0	332.08	0.01776	4.171	4.189	302.76	884.9	1187.6	0.4798	1.1177	1.5974	106.0
108.0	333.44	0.01777	4.097	4.115	304.19	883.8	1188.0	0.4816	1.1144	1.5959	108.0
110.0	334.79	0.01779	4.026	4.044	305.61	882.7	1188.3	0.4834	1.1111	1.5944	110.0
112.0	336.12	0.01780	3.958	3.976	307.00	881.6	1188.6	0.4851	1.1079	1.5930	112.0
114.0	337.43	0.01782	3.892	3.910	308.36	880.6	1188.9	0.4868	1.1048	1.5915	114.0
116.0	338.72	0.01783	3.828	3.846	309.71	879.5	1189.2	0.4885	1.1017	1.5901	116.0
118.0	340.01	0.01785	3.766	3.784	311.05	878.5	1189.5	0.4901	1.0986	1.5887	118.0
120.0	341.26	0.01786	3.707	3.725	312.37	877.4	1189.8	0.4918	1.0956	1.5874	120.0
122.0	342.50	0.01788	3.652	3.670	313.67	876.4	1190.1	0.4934	1.0926	1.5860	122.0
124.0	343.73	0.01789	3.597	3.615	314.96	875.4	1190.4	0.4950	1.0897	1.5847	124.0
126.0	344.94	0.01791	3.542	3.560	316.23	874.4	1190.6	0.4965	1.0868	1.5834	126.0
128.0	346.14	0.01792	3.487	3.505	317.49	873.4	1190.9	0.4981	1.0840	1.5821	128.0
130.0	347.31	0.01794	3.433	3.451	318.73	872.4	1191.2	0.4996	1.0812	1.5808	130.0
132.0	348.48	0.01795	3.383	3.401	319.95	871.5	1191.4	0.5011	1.0784	1.5796	132.0
134.0	349.64	0.01796	3.335	3.353	321.17	870.5	1191.7	0.5026	1.0757	1.5783	134.0
136.0	350.78	0.01798	3.288	3.306	322.37	869.6	1191.9	0.5041	1.0730	1.5771	136.0
138.0	351.91	0.01799	3.242	3.260	323.56	868.6	1192.2	0.5056	1.0703	1.5759	138.0
140.0	353.03	0.01801	3.198	3.216	324.74	867.7	1192.4	0.5070	1.0677	1.5747	140.0
142.0	354.14	0.01802	3.155	3.173	325.91	866.7	1192.6	0.5084	1.0651	1.5735	142.0
144.0	355.22	0.01804	3.112	3.130	327.06	865.8	1192.9	0.5098	1.0625	1.5724	144.0
146.0	356.31	0.01805	3.071	3.089	328.20	864.9	1193.1	0.5112	1.0600	1.5712	146.0
148.0	357.37	0.01806	3.031	3.049	329.32	864.0	1193.3	0.5126	1.0575	1.5701	148.0
150.0	358.43	0.01808	2.992	3.010	330.44	863.1	1193.5	0.5140	1.0550	1.5690	150.0
152.0	359.47	0.01809	2.954	2.972	331.54	862.2	1193.7	0.5153	1.0526	1.5679	152.0
154.0	360.51	0.01810	2.917	2.935	332.64	861.3	1193.9	0.5166	1.0502	1.5668	154.0
156.0	361.53	0.01812	2.882	2.900	333.72	860.4	1194.1	0.5180	1.0478	1.5658	156.0
158.0	362.54	0.01813	2.846	2.864	334.80	859.5	1194.3	0.5193	1.0454	1.5647	158.0
160.0	363.55	0.01814	2.812	2.830	335.86	858.7	1194.5	0.5205	1.0431	1.5636	160.0
162.0	364.54	0.01816	2.779	2.797	336.91	857.8	1194.7	0.5218	1.0408	1.5626	162.0
164.0	365.52	0.01817	2.746	2.764	337.95	857.0	1194.9	0.5230	1.0385	1.5616	164.0
166.0	366.50	0.01818	2.715	2.733	338.99	856.1	1195.1	0.5243	1.0363	1.5606	166.0
168.0	367.46	0.01819	2.683	2.701	340.01	855.2	1195.3	0.5255	1.0340	1.5596	168.0
170.0	368.42	0.01821	2.653	2.671	341.03	854.4	1195.4	0.5268	1.0318	1.5586	170.0
172.0	369.37	0.01822	2.623	2.641	342.04	853.6	1195.6	0.5280	1.0296	1.5576	172.0
174.0	370.31	0.01823	2.594	2.612	343.04	852.7	1195.8	0.5292	1.0275	1.5566	174.0
176.0	371.24	0.01825	2.566	2.584	344.03	851.9	1196.0	0.5304	1.0253	1.5557	176.0
178.0	372.16	0.01826	2.538	2.556	345.01	851.1	1196.1	0.5315	1.0232	1.5548	178.0
180.0	373.08	0.01827	2.511	2.529	345.99	850.3	1196.3	0.5327	1.0211	1.5538	180.0
182.0	374.00	0.01828	2.484	2.502	346.97	849.5	1196.4	0.5339	1.0190	1.5529	182.0
184.0	374.90	0.01829	2.458	2.476	347.94	848.6	1196.6	0.5350	1.0169	1.5520	184.0
186.0	375.78	0.01831	2.433	2.451	348.89	847.9	1196.8	0.5362	1.0149	1.5511	186.0
188.0	376.67	0.01832	2.407	2.425	349.83	847.1	1196.9	0.5373	1.0129	1.5502	188.0
190.0	377.55	0.01833	2.383	2.401	350.77	846.3	1197.0	0.5384	1.0109	1.5493	190.0
192.0	378.42	0.01834	2.359	2.377	351.70	845.5	1197.2	0.5395	1.0089	1.5484	192.0
194.0	379.27	0.01835	2.335	2.353	352.61	844.7	1197.3	0.5406	1.0070	1.5475	194.0
196.0	380.13	0.01837	2.312	2.330	353.53	844.0	1197.5	0.5417	1.0050	1.5467	196.0
198.0	380.97	0.01838	2.289	2.307	354.43	843.2	1197.6	0.5427	1.0031	1.5458	198.0
200.0	381.82	0.01839	2.267	2.285	355.33	842.4	1197.8	0.5438	1.0012	1.5450	200.0
205.0	383.89	0.01842	2.213	2.231	357.56	840.5	1198.1	0.5465	0.9964	1.5429	205.0
210.0	385.93	0.01844	2.162	2.180	359.76	838.6	1198.4	0.5491	0.9918	1.5409	210.0
215.0	387.93	0.01847	2.113	2.131	361.91	836.8	1198.7	0.5516	0.9873	1.5389	215.0
220.0	389.89	0.01850	2.066	2.084	364.02	835.0	1199.0	0.5540	0.9829	1.5369	220.0
225.0	391.81	0.01853	2.0208	2.0393	366.10	833.2	1199.3	0.5565	0.9786	1.5350	225.0
230.0	393.70	0.01856	1.9778	1.9964	368.14	831.4	1199.6	0.5588	0.9743	1.5332	230.0
235.0	395.56	0.01859	1.9367	1.9553	370.15	829.7	1199.8	0.5612	0.9702	1.5313	235.0
240.0	397.40	0.01861	1.8970	1.9156	372.13	827.9	1200.1	0.5635	0.9661	1.5295	240.0
245.0	399.20	0.01864	1.8589	1.8775	374.09	826.2	1200.3	0.5658	0.9620	1.5278	245.0

TABLE A-3-2
SATURATED STEAM: TEMPERATURE TABLE

Temp. Fahr. t	Abs. Press. Lb./Sq. In. p	Specific Volume			Enthalpy			Entropy			Temp. Fahr. t
		Sat. Liquid v _l	Evap. v _{fg}	Sat. Vapor v _g	Sat. Liquid h _l	Evap. h _{fg}	Sat. Vapor h _g	Sat. Liquid s _l	Evap. s _{fg}	Sat. Vapor s _g	
32°	0.0887	0.01602	3301	3303	/ 0.00	1073.4	1073.4	0.0000	2.1834	2.1834	32°
34	0.0961	0.01602	3059	3059	2.01	1072.3	1074.3	0.0041	2.1723	2.1764	34
36	0.1041	0.01602	2835	2835	4.03	1071.2	1075.3	0.0081	2.1614	2.1695	36
38	0.1126	0.01602	2632	2632	6.04	1070.2	1076.2	0.0122	2.1505	2.1627	38
40°	0.1217	0.01602	2445	2445	8.05	1069.1	1077.1	0.0162	2.1397	2.1559	40°
42	0.1315	0.01602	2272	2272	10.05	1068.0	1078.0	0.0202	2.1290	2.1492	42
44	0.1420	0.01602	2112	2112	12.05	1066.9	1078.9	0.0242	2.1184	2.1426	44
46	0.1532	0.01602	1965.3	1965.3	14.06	1065.8	1079.8	0.0282	2.1078	2.1360	46
48	0.1652	0.01602	1829.7	1829.7	16.06	1064.7	1080.8	0.0321	2.0974	2.1295	48
50°	0.1780	0.01602	1704.8	1704.8	18.06	1063.6	1081.7	0.0361	2.0870	2.1231	50°
52	0.1918	0.01602	1588.3	1588.3	20.06	1062.5	1082.6	0.0400	2.0767	2.1167	52
54	0.2063	0.01602	1482.4	1482.4	22.06	1061.4	1083.5	0.0439	2.0665	2.1104	54
56	0.2219	0.01603	1383.5	1383.5	24.05	1060.3	1084.4	0.0478	2.0564	2.1042	56
58	0.2384	0.01603	1292.8	1292.8	26.05	1059.3	1085.3	0.0517	2.0464	2.0980	58
60°	0.2561	0.01603	1208.0	1208.0	28.05	1058.2	1086.2	0.0555	2.0364	2.0919	60°
62	0.2749	0.01603	1129.7	1129.7	30.05	1057.1	1087.1	0.0594	2.0265	2.0859	62
64	0.2949	0.01604	1057.1	1057.1	32.04	1056.0	1088.0	0.0632	2.0167	2.0799	64
66	0.3162	0.01604	989.7	989.6	34.04	1054.9	1089.0	0.0670	2.0069	2.0739	66
68	0.3388	0.01605	927.1	927.1	36.03	1053.8	1089.8	0.0708	1.9973	2.0680	68
70°	0.3628	0.01605	869.0	869.0	38.03	1052.7	1090.8	0.0746	1.9877	2.0622	70°
72	0.3883	0.01606	814.9	815.0	40.02	1051.6	1091.7	0.0783	1.9782	2.0565	72
74	0.4153	0.01606	764.8	764.8	42.02	1050.6	1092.6	0.0820	1.9687	2.0507	74
76	0.4440	0.01606	718.0	718.0	44.01	1049.5	1093.5	0.0858	1.9593	2.0451	76
78	0.4744	0.01607	674.5	674.5	46.00	1048.4	1094.4	0.0895	1.9500	2.0395	78
80°	0.5067	0.01607	633.8	633.8	48.00	1047.3	1095.3	0.0932	1.9407	2.0340	80°
82	0.5409	0.01608	595.9	595.9	50.00	1046.2	1096.2	0.0969	1.9316	2.0285	82
84	0.5772	0.01608	560.5	560.5	52.00	1045.1	1097.1	0.1006	1.9224	2.0230	84
86	0.6153	0.01609	527.6	527.7	54.00	1044.0	1098.0	0.1042	1.9134	2.0176	86
88	0.6555	0.01609	497.1	497.1	56.00	1042.9	1098.9	0.1079	1.9044	2.0123	88
90°	0.6980	0.01610	468.5	468.5	58.00	1041.8	1099.8	0.1115	1.8955	2.0070	90°
92	0.7429	0.01610	441.7	441.8	59.98	1040.7	1100.7	0.1152	1.8866	2.0018	92
94	0.7902	0.01611	416.8	416.8	61.97	1039.6	1101.6	0.1188	1.8778	1.9966	94
96	0.8403	0.01612	393.3	393.3	63.96	1038.5	1102.5	0.1224	1.8691	1.9915	96
98	0.8930	0.01612	371.4	371.4	65.94	1037.4	1103.4	0.1260	1.8604	1.9864	98
100°	0.9487	0.01613	350.8	350.8	67.93	1036.3	1104.2	0.1295	1.8518	1.9813	100°
105	1.1009	0.01615	305.0	305.0	72.91	1033.5	1106.4	0.1384	1.8305	1.9689	105
110	1.274	0.01616	265.8	265.8	77.89	1030.8	1108.6	0.1472	1.8095	1.9567	110
115	1.470	0.01618	232.3	232.3	82.89	1027.9	1110.8	0.1559	1.7889	1.9448	115
120°	1.692	0.01620	203.5	203.6	87.88	1025.1	1113.0	0.1646	1.7685	1.9331	120°
125	1.941	0.01622	178.9	178.9	92.87	1022.2	1115.1	0.1731	1.7485	1.9216	125
130	2.221	0.01625	157.62	157.64	97.86	1019.4	1117.2	0.1816	1.7288	1.9104	130
135	2.536	0.01627	139.15	139.17	102.85	1016.5	1119.3	0.1901	1.7094	1.8995	135
140	2.887	0.01629	123.20	123.22	107.84	1013.6	1121.4	0.1984	1.6903	1.8887	140
145°	3.280	0.01632	109.29	109.31	112.84	1010.6	1123.5	0.2067	1.6715	1.8782	145°
150	3.716	0.01634	97.21	97.23	117.84	1007.7	1125.5	0.2149	1.6530	1.8679	150
155	4.201	0.01637	86.64	86.66	122.85	1004.7	1127.6	0.2231	1.6347	1.8578	155
160	4.739	0.01639	77.38	77.40	127.85	1001.8	1129.6	0.2312	1.6168	1.8479	160
165	5.334	0.01642	69.26	69.28	132.85	998.8	1131.7	0.2392	1.5991	1.8383	165
170°	5.990	0.01645	62.13	62.14	137.85	995.8	1133.7	0.2472	1.5816	1.8288	170°
175	6.716	0.01648	55.81	55.82	142.86	992.8	1135.7	0.2551	1.5644	1.8195	175
180	7.510	0.01650	50.26	50.28	147.87	989.8	1137.7	0.2629	1.5475	1.8105	180
185	8.382	0.01654	45.35	45.36	152.87	986.8	1139.7	0.2707	1.5308	1.8016	185
190	9.336	0.01656	40.99	41.01	157.89	983.8	1141.7	0.2785	1.5144	1.7929	190
195°	10.385	0.01660	37.10	37.12	162.91	980.8	1143.7	0.2862	1.4982	1.7844	195°
200	11.525	0.01663	33.65	33.67	167.94	977.7	1145.6	0.2938	1.4822	1.7760	200
205	12.769	0.01665	30.57	30.59	172.97	974.6	1147.6	0.3014	1.4664	1.7678	205
210	14.123	0.01669	27.82	27.83	177.99	971.5	1149.5	0.3089	1.4508	1.7597	210
215	14.696	0.01670	26.80	26.82	180.00	970.2	1150.2	0.3119	1.4446	1.7564	215
215°	15.591	0.01673	25.35	25.37	183.02	968.3	1151.3	0.3164	1.4353	1.7517	215°
220	17.188	0.01676	23.14	23.16	188.06	965.1	1153.1	0.3238	1.4200	1.7439	220
225	18.915	0.01680	21.15	21.17	193.09	961.8	1154.9	0.3313	1.4049	1.7362	225
230	20.78	0.01683	19.371	19.388	198.15	958.6	1156.7	0.3386	1.3900	1.7286	230
235	22.80	0.01687	17.761	17.778	203.21	955.3	1158.5	0.3460	1.3752	1.7212	235

TABLE A-3-3
SUPERHEATED STEAM

Abs. Press. Lb./Sq. In. (Sat. Temp.)		Sat. Water	Sat. Steam	Temperature—Degrees Fahrenheit															
				160°	200°	250°	300°	350°	400°	450°	500°	550°	600°	700°	800°	900°	1000°		
1 (101.76)	v	0.02	333.9	362.7	392.5	422.3	452.1	481.9	511.7	541.6	571.3	601.1	630.9	690.6	750.2	809.8	869.4		
	h	69.7	1105.0	1127.0	1149.8	1172.4	1195.0	1217.8	1240.7	1263.9	1287.2	1310.9	1334.6	1383.0	1432.6	1483.3	1535.2		
5 (162.25)	v	0.1326	1.9769	2.0144	2.0503	2.0835	2.1142	2.1433	2.1707	2.1968	2.2218	2.2458	2.2688	2.3125	2.3535	2.3922	2.4291		
	h	0.02	73.61	78.17	84.19	90.21	96.21	102.19	108.19	114.16	120.14	126.11	132.09	138.05	143.99	149.91	155.83		
10 (193.21)	v	0.2348	1.8435	1.8712	1.9050	1.9361	1.9654	1.9930	2.0192	2.0443	2.0683	2.0914	2.1135	1.82.9	1.84.2	1.85.5	1.86.8		
	h	0.02	38.45	38.88	41.92	44.98	48.00	51.01	54.02	57.02	60.02	63.01	65.99	74.96	80.92	86.89	92.86		
14.698 (212.00)	v	0.2834	1.7874	1.146.3	1.169.9	1.193.2	1.216.4	1.239.6	1.262.9	1.286.4	1.310.1	1.334.0	1.357.2	1.382.4	1.432.1	1.482.9	1.534.9		
	h	0.02	26.82	28.40	30.52	32.59	34.65	36.70	38.75	40.79	42.83	44.86	46.91	50.97	55.03	59.09	63.15		
20 (227.96)	v	0.3119	1.7564	1.7832	1.8154	1.8452	1.8731	1.8996	1.9249	1.9490	1.9722	1.9944	2.0161	2.0572	2.0961	2.1330	2.1688		
	h	0.02	20.10	20.79	22.36	23.90	25.43	26.94	28.45	29.95	31.46	32.95	34.46	37.44	40.43	43.42	46.41		
40 (267.24)	v	0.017	10.497	11.044	11.843	12.623	13.392	14.161	14.922	15.682	16.438	17.190	17.938	18.686	19.428	20.168	20.907		
	h	0.2359	1169.2	1186.5	1211.3	1235.6	1259.7	1283.7	1307.8	1332.0	1356.2	1380.2	1404.1	1428.0	1451.9	1475.8	1500.0		
60 (292.71)	v	0.017	10.497	11.044	11.843	12.623	13.392	14.161	14.922	15.682	16.438	17.190	17.938	18.686	19.428	20.168	20.907		
	h	0.2620	1177.0	1188.2	1213.8	1238.3	1262.9	1287.6	1312.4	1337.2	1362.0	1386.8	1411.6	1436.4	1461.2	1486.0	1510.8		
80 (312.03)	v	0.018	5.470	5.806	6.217	6.618	7.015	7.406	7.793	8.176	8.558	8.933	9.313	10.067	10.817	11.565	12.313		
	h	0.018	5.470	5.806	6.217	6.618	7.015	7.406	7.793	8.176	8.558	8.933	9.313	10.067	10.817	11.565	12.313		
100 (327.83)	v	0.018	4.426	4.594	4.934	5.263	5.585	5.903	6.215	6.528	6.831	7.139	7.446	8.044	8.644	9.244	9.844		
	h	0.018	4.426	4.594	4.934	5.263	5.585	5.903	6.215	6.528	6.831	7.139	7.446	8.044	8.644	9.244	9.844		
120 (341.26)	v	0.018	3.725	3.781	4.077	4.359	4.632	4.899	5.162	5.425	5.688	5.951	6.214	6.819	7.424	8.029	8.634		
	h	0.018	3.725	3.781	4.077	4.359	4.632	4.899	5.162	5.425	5.688	5.951	6.214	6.819	7.424	8.029	8.634		
140 (353.03)	v	0.018	3.216	3.465	3.713	3.951	4.182	4.410	4.638	4.866	5.094	5.322	5.550	6.155	6.760	7.365	7.970		
	h	0.018	3.216	3.465	3.713	3.951	4.182	4.410	4.638	4.866	5.094	5.322	5.550	6.155	6.760	7.365	7.970		
160 (363.55)	v	0.018	2.830	3.005	3.227	3.440	3.645	3.846	4.046	4.246	4.446	4.646	4.846	5.451	6.056	6.661	7.266		
	h	0.018	2.830	3.005	3.227	3.440	3.645	3.846	4.046	4.246	4.446	4.646	4.846	5.451	6.056	6.661	7.266		
180 (373.08)	v	0.018	2.529	2.646	2.849	3.041	3.226	3.407	3.588	3.769	3.950	4.131	4.312	4.917	5.522	6.127	6.732		
	h	0.018	2.529	2.646	2.849	3.041	3.226	3.407	3.588	3.769	3.950	4.131	4.312	4.917	5.522	6.127	6.732		
200 (381.82)	v	0.018	2.285	2.358	2.547	2.722	2.892	3.056	3.220	3.384	3.548	3.712	3.876	4.481	5.086	5.691	6.296		
	h	0.018	2.285	2.358	2.547	2.722	2.892	3.056	3.220	3.384	3.548	3.712	3.876	4.481	5.086	5.691	6.296		
220 (389.89)	v	0.019	2.084	2.122	2.299	2.462	2.617	2.769	2.921	3.073	3.225	3.377	3.529	4.134	4.739	5.344	5.949		
	h	0.019	2.084	2.122	2.299	2.462	2.617	2.769	2.921	3.073	3.225	3.377	3.529	4.134	4.739	5.344	5.949		
240 (397.40)	v	0.0186	1.9156	1.9250	2.092	2.244	2.389	2.529	2.670	2.811	2.952	3.093	3.234	3.839	4.444	5.049	5.654		
	h	0.0186	1.9156	1.9250	2.092	2.244	2.389	2.529	2.670	2.811	2.952	3.093	3.234	3.839	4.444	5.049	5.654		
260 (404.43)	v	0.0187	1.7723	1.9165	2.062	2.195	2.327	2.459	2.591	2.723	2.855	2.987	3.119	3.724	4.329	4.934	5.539		
	h	0.0187	1.7723	1.9165	2.062	2.195	2.327	2.459	2.591	2.723	2.855	2.987	3.119	3.724	4.329	4.934	5.539		
280 (411.06)	v	0.0188	1.6490	1.7655	1.9019	2.030	2.153	2.276	2.400	2.523	2.646	2.769	2.892	3.497	4.102	4.707	5.312		
	h	0.0188	1.6490	1.7655	1.9019	2.030	2.153	2.276	2.400	2.523	2.646	2.769	2.892	3.497	4.102	4.707	5.312		
300 (417.33)	v	0.0189	1.5414	1.6347	1.7648	1.8863	2.002	2.124	2.246	2.368	2.490	2.612	2.734	3.339	3.944	4.549	5.154		
	h	0.0189	1.5414	1.6347	1.7648	1.8863	2.002	2.124	2.246	2.368	2.490	2.612	2.734	3.339	3.944	4.549	5.154		
350 (431.71)	v	0.0191	1.3245	1.3712	1.4899	1.5981	1.7003	1.7975	1.8947	1.9919	2.0891	2.1863	2.2835	2.888	3.493	4.098	4.703		
	h	0.0191	1.3245	1.3712	1.4899	1.5981	1.7003	1.7975	1.8947	1.9919	2.0891	2.1863	2.2835	2.888	3.493	4.098	4.703		
400 (444.58)	v	0.0194	1.1601	1.1712	1.2828	1.3817	1.4740	1.5663	1.6586	1.7509	1.8432	1.9355	2.0278	2.632	3.237	3.842	4.447		
	h	0.0194	1.1601	1.1712	1.2828	1.3817	1.4740	1.5663	1.6586	1.7509	1.8432	1.9355	2.0278	2.632	3.237	3.842	4.447		

TABLE A-4
EQUIVALENTS OF UNITS

1 lb	= 0.4536 kg	1 kg	= 2.2046 lb
	= 7000 grains	1 kg	= 15.43×10^3 grains
1 ft	= 0.3048 m	1 m	= 39.37 in.
1 in.	= 2.54 cm	1 cm	= 0.394 in.
1 sq ft	= 0.09290 sq m	1 sq m	= 10.7639 sq ft
1 cu ft	= 0.0283 cu m	1 cu m	= 35.315 cu ft
1 deg F	= $\frac{5}{9}$ deg C	1 deg C	= 1.8 deg F
Deg F	= 1.8 deg C + 32°	Deg C	= $\frac{5}{9}(\text{deg F} - 32)$
Abs deg F	= Deg F + 459.6	Abs deg C	= deg C + 273.18
	= deg Rankine		= deg Kelvin
1 atm	= 14.696 lb per sq in.	1 atm	= 1.033 kg per sq cm
	= 29.921 in. Hg		= 760 mm Hg
	= 33.90 ft water at 39.1° F		= 10.33 m water at 4° C
1 lb per sq in.	= 0.0703 kg per sq cm	1 kg per sq cm	= 14.223 lb per sq in.
1 in. water	= 0.036 lb per sq in.	1 oz per sq in.	= 1.72 in. water
1 in. water	= 0.58 oz per sq in.	1 oz per sq in.	= 0.0127 in. Hg
1 in. Hg	= 0.491 lb per sq in.	1 in. Hg	= 13.595 in. water
1 U. S. gal	= 231 cu in.	1 liter	= 0.264 U. S. gal
1 Btu	= 778.6 ft-lb	1 calorie	= 427.16 kg-m
1 Btu	= 0.252 calorie	1 calorie	= 3.968 Btu
1 watt-hr	= 3.410 Btu	1 watt-hr	= 0.859 calorie
1 watt-hr	= 2.655×10^3 ft-lb	1 watt-hr	= 367.10 kg-m
1 hp-hr	= 745.7 watts	1 watt	= 0.00134 hp
1 ft-lb	= 0.1383 kg-m	1 kg m	= 7.233 ft-lb
1 ft-lb	= 1.284×10^{-3} Btu	1 kg m	= 2.34×10^{-3} calorie
1 ft-lb	= 3.24×10^{-2} calorie	1 kg m	= 9.29×10^{-3} Btu
<i>g</i>	= 32.174 ft per sec ²	<i>g</i>	= 980.7 cm per sec ²

TABLE A-5
INFILTRATION THROUGH WINDOWS *

Expressed in Cubic Feet per Foot of Crack per Hour(a)

Type of Window	Remarks	Wind Velocity, Miles per Hour					
		5	10	15	20	25	30
Double-Hung Wood Sash Windows (Unlocked)	Around frame in masonry wall—not calked(b).....	3.3	8.2	14.0	20.2	27.2	34.6
	Around frame in masonry wall—calked(b).....	0.5	1.5	2.6	3.8	4.8	5.8
	Around frame in wood frame construction(b).....	2.2	6.2	10.8	16.6	23.0	30.3
	Total for average window, non-weatherstripped, $\frac{1}{8}$ -in. crack and $\frac{1}{4}$ -in. clearance(c). Includes wood frame leakage(d).....	6.6	21.4	39.3	59.3	80.0	103.7
	Ditto, weatherstripped(d).....	4.3	15.5	23.6	35.5	48.6	63.4
	Total for poorly fitted window, non-weatherstripped, $\frac{1}{8}$ -in. crack and $\frac{1}{4}$ -in. clearance(e). Includes wood frame leakage(f).....	26.9	69.0	110.5	153.9	199.2	249.4
	Ditto, weatherstripped(d).....	5.9	18.9	34.1	51.4	70.5	91.5
Double-Hung Metal Windows(f)	Non-weatherstripped, locked.....	20	45	70	96	125	154
	Non-weatherstripped, unlocked.....	20	47	74	104	137	170
	Weatherstripped, unlocked.....	6	19	32	46	60	76
Rolled Section Steel Sash Windows(k)	Industrial pivoted(g), $\frac{1}{8}$ -in. crack..	52	108	176	244	304	372
	Architectural projected(h), $\frac{1}{8}$ -in. crack..	20	52	88	116	152	208
	Residential, casement(i), $\frac{1}{8}$ -in. crack..	14	32	52	76	100	128
	Heavy casement section, projected(j), $\frac{1}{8}$ -in. crack.....	8	24	38	54	72	96
Hollow metal, vertically pivoted window(f).....		30	88	145	186	221	242

* Reproduced by permission from 1935 Guide A S.H.V.E.

(a) The values given in this table are 20 per cent less than test values to allow for building up of pressure in rooms.

(b) The values given for frame leakage are per foot of sash perimeter as determined for double hung wood windows. Some of the frame leakage in masonry walls originates in the brick wall itself and cannot be prevented by calking. For the additional reason that calking is not done perfectly and deteriorates with time, it is considered advisable to choose the masonry frame leakage values for calked frames as the average determined by the calked and not-calked tests.

(c) The fit of the average double-hung wood window was determined as $\frac{1}{8}$ -in. crack and $\frac{1}{4}$ -in. clearance by measurements on approximately 600 windows under heating season conditions.

(d) The values given are the totals for the window opening per foot of sash perimeter and include frame leakage and so-called *elsewhere* leakage. The frame leakage values included are for wood frame construction but apply as well to masonry construction, assuming a 50 per cent efficiency of frame calking.

(e) A $\frac{1}{8}$ -in. crack and clearance represents a poorly fitted window, much poorer than average.

(f) Windows tested in place in building.

(g) Industrial pivoted window generally used in industrial buildings. Ventilators horizontally pivoted at center or slightly above, lower part swinging out.

(h) Architectural projected made of same sections as industrial pivoted except that outside framing member is heavier, and refinements in weathering and hardware. Used in semi-monumental buildings such as schools. Ventilators swing in or out and are balanced on side arms.

(i) Of same design and section shapes as so-called *heavy section casement* but of lighter weight.

(j) Made of heavy sections. Ventilators swing in or out and stay set at any degree of opening.

(k) With reasonable care in installation, leakage at contacts where windows are attached to steel framework and at mullions is negligible. With $\frac{1}{8}$ -in. crack, representing poor installation, leakage at contact with steel framework is about one-third, and at mullions about one-sixth of that given for industrial pivoted windows in the table.

APPENDIX

TABLE A-6
RADIATOR SPACE REQUIREMENTS

Courtesy of Crane Co.

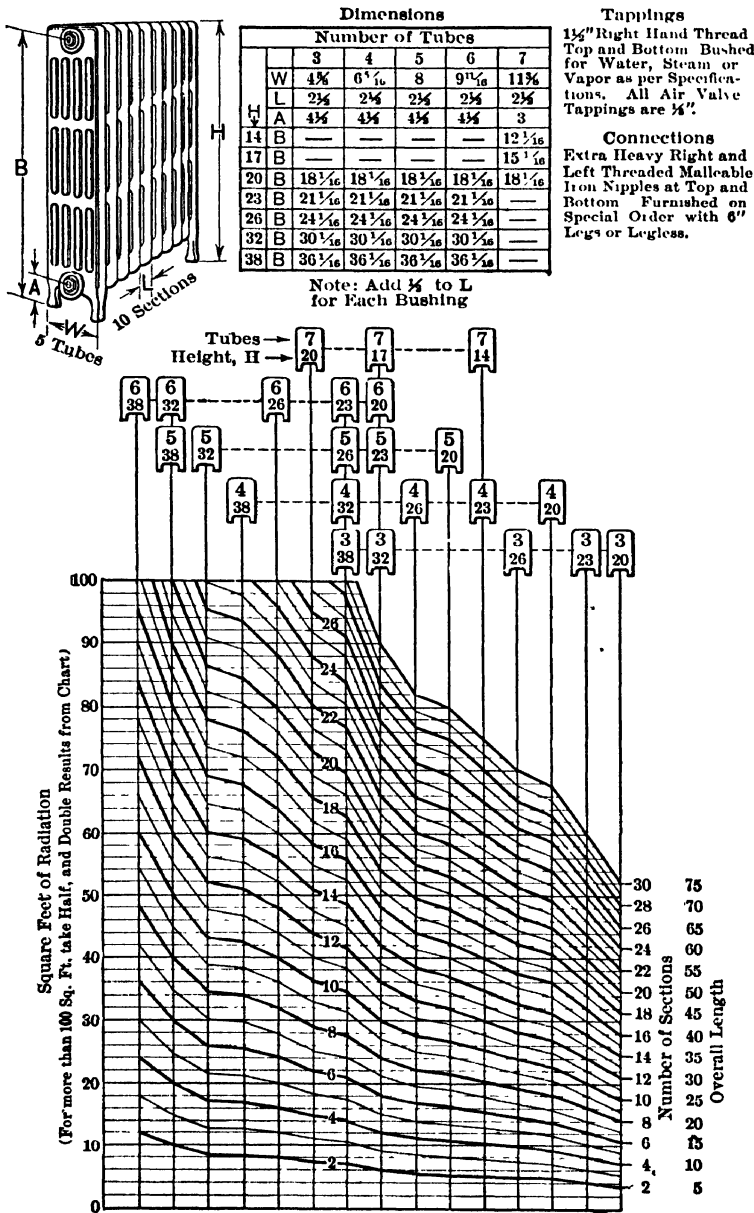


TABLE A-9
CONVERSION FACTORS FOR DIRECT CAST-IRON RADIATORS *

Steam Pressure	Temper- ature of Steam or Water, deg F	Temperature of Room					
		80° F	75° F	70° F	65° F	60° F	50° F
3.71	150	0.388	0.424	0.462	0.499	0.538	0.617
4.75	160	0.462	0.499	0.538	0.577	0.617	0.697
5.99	170	0.538	0.577	0.617	0.657	0.697	0.782
7.51	180	0.617	0.657	0.697	0.740	0.782	0.868
9.33	190	0.697	0.740	0.782	0.825	0.868	0.955
11.52	200	0.782	0.825	0.868	0.911	0.955	1.045
15.7	215	0.911	0.955	1.000	1.045	1.091	1.183
16.7	219	0.947	0.991	1.036	1.081	1.128	1.220
17.7	222	0.973	1.018	1.063	1.109	1.155	1.249
18.7	225	1.000	1.045	1.091	1.137	1.183	1.277
19.7	227	1.018	1.063	1.109	1.155	1.202	1.296
20.7	230	1.045	1.091	1.137	1.183	1.230	1.325
22.7	235	1.091	1.137	1.183	1.230	1.277	1.373
24.7	240	1.137	1.183	1.230	1.277	1.325	1.421

* From A.S.H.V.E. Guide, 1933, by permission.

TABLE A-10
 DIMENSIONAL DATA NOS. 61, 71, AND 72 FLEXITUBE AEROFIN HEATERS. 29" FACE WITH 18 TUBES
Courtesy of Aerofin Corporation

Length of Tubes, Feet	Net Face Area,* Sq Ft		Inside Dimensions Over Fin Surface		Overall Dimensions of Casing		Sq Ft of Heating Surface			Weight in Pounds Including Casing		
	Nos. 61 and 71	No. 72	Across Tubes		Across Tubes	Along Tubes	No. 61	No. 71	No. 72	No. 61	No. 71	No. 72
			Nos. 61 and 71	No. 72								
2	4.09	4.24	25 $\frac{1}{8}$ "	26"	23 $\frac{1}{8}$ "	29"	40	57	113	70	80	103
2 $\frac{1}{2}$	5.14	5.33	25 $\frac{1}{8}$ "	26"	29 $\frac{1}{2}$ "	29"	49	70	140	77	90	117
3	6.18	6.41	25 $\frac{1}{8}$ "	26"	35 $\frac{1}{2}$ "	29"	58	84	167	83	98	130
3 $\frac{1}{2}$	7.22	7.50	25 $\frac{1}{8}$ "	26"	41 $\frac{1}{2}$ "	29"	68	97	194	101	120	151
4	8.27	8.58	25 $\frac{1}{8}$ "	26"	47 $\frac{1}{2}$ "	29"	77	111	221	108	127	163
4 $\frac{1}{2}$	9.31	9.66	25 $\frac{1}{8}$ "	26"	53 $\frac{1}{2}$ "	29"	86	124	248	118	140	181
5	10.36	10.73	25 $\frac{1}{8}$ "	26"	59 $\frac{1}{2}$ "	29"	96	138	275	124	149	194
5 $\frac{1}{2}$	11.40	11.82	25 $\frac{1}{8}$ "	26"	65 $\frac{1}{2}$ "	29"	105	151	302	129	157	211
6	12.45	12.90	25 $\frac{1}{8}$ "	26"	71 $\frac{1}{2}$ "	29"	115	165	329	139	170	228
7	14.53	15.08	25 $\frac{1}{8}$ "	26"	83 $\frac{1}{2}$ "	29"	133	192	383	148	185	259
8	16.62	17.24	25 $\frac{1}{8}$ "	26"	95 $\frac{1}{2}$ "	29"	152	219	437	171	213	290
9	18.71	19.41	25 $\frac{1}{8}$ "	26"	107 $\frac{1}{2}$ "	29"	171	246	491	184	231	327
10	20.80	21.58	25 $\frac{1}{8}$ "	26"	119 $\frac{1}{2}$ "	29"	190	273	545	242	294	399

* Net Face Area means only that area facing the tubes and does not include the headers or casings.
 Note.—Depth of casings of all standard units is 10 in. in the direction of air flow. Depth of casings of all narrow units is 5 in. in the direction of air flow.

TABLE A-11
FINAL TEMPERATURE AND CONDENSATION PERFORMANCE OF
Nos. 61, 71 AND 72 FLEXITUBE AEROFIN HEATERS

Steam at 227° F and velocity of air at 70° F and 29.92 in. in ft per min measured through net face area facing tubes as given in Table A-10

Courtesy of Aerofin Corporation
No. 61 FLEXITUBE

Temperature Entering Air	Units Deep	300' Face Velocity		400' Face Velocity		500' Face Velocity		600' Face Velocity		700' Face Velocity	
		Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.
-20°	1	27.3	1.71	22.5	2.05	19.2	2.36	16.8	2.66	14.7	2.93
	2	69.2	1.66	61.2	1.96	55.4	2.27	50.9	2.56	47.4	2.85
-10°	1	35.4	1.64	30.8	1.97	27.5	2.26	25.2	2.54	23.3	2.81
	2	73.6	1.51	65.4	1.82	59.8	2.10	56.3	2.40	53.7	2.69
0°	1	43.5	1.57	39.1	1.88	36.0	2.17	33.8	2.44	31.9	2.69
	2	78.9	1.43	71.0	1.71	65.8	1.98	62.6	2.26	59.8	2.52
+10°	1	51.6	1.50	47.4	1.80	44.4	2.07	42.3	2.34	40.5	2.57
+20°	1	59.6	1.43	55.6	1.72	52.8	1.98	50.8	2.23	49.1	2.45
+30°	1	67.8	1.37	63.9	1.64	61.2	1.88	59.3	2.12	57.7	2.34
+40°	1	75.8	1.29	72.2	1.56	69.6	1.78	67.8	2.01	66.3	2.22
+50°	1	83.9	1.23	80.5	1.47	78.1	1.70	76.4	1.91	74.9	2.10
+60°	1	92.0	1.16	88.8	1.39	86.5	1.60	84.9	1.80	83.5	1.98
+70°	1	100.1	1.09	97.0	1.30	94.9	1.50	93.4	1.69	92.1	1.86

Nos. 71 AND 72 FLEXITUBE

Temperature Entering Air	Rows Tubes Deep	300' Face Velocity		400' Face Velocity		500' Face Velocity		600' Face Velocity		700' Face Velocity	
		Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.	Final Temperature Air	Conden- sation, Lb per Hr per Sq Ft H. S.
-20°	1	48.8	1.71	41.7	2.04	37.1	2.36	34.1	2.68	31.5	2.98
	2	98.5	1.47	89.5	1.81	81.5	2.10	76.0	2.38	71.8	2.66
	3	134.1	1.27	124.6	1.59	116.0	1.87	109.4	2.14	103.7	2.39
	4	159.8	1.12	150.8	1.41	142.0	1.67	135.0	1.92	129.0	2.16
	5	178.6	0.99	169.8	1.26	162.0	1.51	154.9	1.74	149.1	1.96
	6	192.4	0.88	184.5	1.13	177.0	1.36	170.6	1.58	165.0	1.78
0°	1	63.2	1.57	56.7	1.87	52.4	2.17	49.7	2.47	47.3	2.74
	2	108.9	1.35	100.6	1.66	93.2	1.93	88.1	2.19	84.4	2.44
	3	141.6	1.17	132.8	1.46	124.9	1.72	118.9	1.97	113.6	2.19
	4	165.2	1.02	156.9	1.30	148.9	1.54	142.4	1.77	136.9	1.98
	5	182.5	0.91	174.4	1.15	167.2	1.38	160.7	1.59	155.3	1.80
	6	195.2	0.81	187.9	1.04	181.1	1.25	175.2	1.45	170.1	1.64
+30°	1	84.8	1.36	79.2	1.63	75.5	1.88	73.1	2.14	71.0	2.37
	2	124.6	1.17	117.4	1.45	111.0	1.67	106.5	1.90	103.3	2.12
	3	152.9	1.02	145.3	1.27	138.4	1.49	133.2	1.71	128.6	1.90
	4	173.4	0.89	166.2	1.13	159.3	1.34	153.6	1.53	148.8	1.72
+40°	1	92.1	1.29	86.7	1.54	83.2	1.79	81.0	2.03	79.0	2.26
	2	129.7	1.11	122.9	1.37	116.8	1.59	111.6	1.78	109.6	2.01
	3	156.7	0.97	149.4	1.21	142.9	1.42	137.9	1.62	133.7	1.81
	4	176.1	0.84	169.3	1.07	162.6	1.27	157.3	1.46	152.8	1.63
+60°	1	106.5	1.15	101.7	1.38	98.6	1.60	96.6	1.82	94.8	2.01
	2	140.1	0.99	134.1	1.23	128.6	1.42	124.9	1.61	121.1	1.77
	3	164.3	0.86	158.2	1.08	151.9	1.27	147.5	1.45	143.7	1.61
	4	181.6	0.75	175.4	0.95	169.5	1.13	164.8	1.30	160.7	1.46

TABLE A-12

**AIR FRICTION LOSS IN FLEXITUBE AEROFIN UNITS AT 70° F AND
29.92 IN. IN INCHES OF WATER**

Courtesy of Aerofin Corporation

Units	Rows of Tubes	Velocity of 70° Air in Ft per Min through Net Face Area					
		300	400	500	600	700	800
No. 61	1	0.020	0.037	0.061	0.090	0.127	0.170
	2	0.038	0.068	0.106	0.154	0.210	0.276
No. 71 and No. 72	1	0.030	0.050	0.074	0.103	0.136	0.174
	2	0.044	0.076	0.116	0.164	0.221	0.285
	3	0.058	0.101	0.158	0.225	0.306	0.396
	4	0.072	0.127	0.200	0.286	0.391	0.507
	6	0.101	0.178	0.283	0.408	0.561	0.729
	8	0.130	0.230	0.367	0.530	0.731	0.951
	10	0.158	0.281	0.450	0.652	0.901	1.173

TABLE A-13

DIMENSIONS OF REGULAR VENTO SECTION

Courtesy of American Radiator Company

Nominal Size, in.	Sq ft Surface	Height, in.	Width, in.	Thick-ness, in.	Sq Ft Free Air Between Sections at Different Center Distances			
					5 $\frac{3}{8}$ in.	5 in.	4 $\frac{3}{8}$ in.	4 in.
30	8.00	29 $\frac{7}{8}$	9 $\frac{1}{8}$	3 $\frac{7}{32}$	0.541	0.460	0.390	0.255
40	10.75	40 $\frac{1}{8}$	9 $\frac{1}{8}$	3 $\frac{7}{32}$	0.729	0.620	0.525	0.350
50	13.50	50 $\frac{3}{4}$	9 $\frac{1}{8}$	3 $\frac{7}{32}$	0.902	0.726	0.650	0.428
60	16.00	60 $\frac{1}{4}$	9 $\frac{1}{8}$	3 $\frac{7}{32}$	1.083	0.921	0.781	0.511
72	19.00	72 $\frac{3}{4}$	9 $\frac{1}{8}$	3 $\frac{7}{32}$	1.31	1.104	0.937	

TABLE A-14
VENTO HEATER DATA

REGULAR SECTION—RATINGS AND FREE AREAS

Courtesy of American Radiator Company

60 in. Section (Steam or Water)—16 Sq Ft. Height $60\frac{1}{8}$ in. Width $9\frac{1}{8}$ in.

Number of Sections in Stack	Square Feet of Heating Surface	*Equiv- alent in Lineal Feet 1 in. Pipe	5 $\frac{3}{8}$ in Centers of Sections		5 in. Centers of Sections		4 $\frac{5}{8}$ in. Centers of Sections		Actual Weights
			52% of Face		Standard 44% of Face		37% of Face		
			Net Air Space in Square Feet	†Width of Stack in Inches	Net Air Space in Square Feet	†Width of Stack in Inches	Net Air Space in Square Feet	†Width of Stack in Inches	
4	64 0	192	4 38	21	3 69	20	3.13	18	8.20 lb per sq ft actual 9 lb per sq ft shipping weight
5	80.0	240	5 46	27	4 61	25	3.91	23	
6	96 0	288	6 54	32	5 53	30	4.69	28	
7	112.0	336	7 62	38	6 45	35	5.47	32	
8	128.0	384	8 70	43	7.37	40	6.25	37	
9	144.0	432	9 77	48	8 29	45	7.03	42	
10	160 0	480	10 85	54	9 21	50	7.81	46	
12	192.0	576	13.00	65	11 05	60	9.37	55	
14	224.0	672	15.15	75	12 89	70	10.93	65	
15	240 0	720	16.23	81	13 81	75	11.71	69	
16	256.0	768	17.31	86	14 73	80	12.49	74	
18	288.0	864	19 46	97	16 57	90	14.05	83	
20	320.0	960	21.62	108	18 42	100	15.61	92	
22	352.0	1056	23.78	118	20.26	110	17.17	102	
24	384.0	1152	25 93	129	22.10	120	18.73	111	

NOTE.—60-inch sections can be assembled on 4 in. centers (25–27).

* The actual length of 1 in. pipe per square foot of outside surface is 2.9 lineal feet but is nominally figured at 3 lineal feet, as shown in the third column of above table.

† Add to the width of stack $2\frac{1}{8}$ in. for staggering of stacks—except 4 in. centers not staggered.

TABLE A-15
FINAL TEMPERATURE AND CONDENSATION PERFORMANCE OF REGULAR SECTION VENTO HEATERS WITH
5 IN. CENTERS

Courtesy of American Radiator Company

Steam at 227° F and velocities of air at 70° F and 20.92 in. in feet per minute measured through net area between sections as given in Table A-14.

Velocity	100		120		200		600		800		1000		1200		1400		1600		1800	
	Num- ber of Stacks Deep	Temp- ature of Entering Air	Final Temp. Air Leaving Heater	Cond. Lb per Sq Ft per Hr	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.	F. T.	C.
1	0	76	0.49	73	0.56	62	0.79	65	38	1.95	35	2.24	32	2.46	47	2.42	45	2.56	43	2.65
	20	87	.43	83	.48	75	.70	58	54	1.75	51	1.99	49	2.23	56	2.33	54	2.46	52	2.54
	30	93	.40	89	.45	81	.65	66	62	1.64	60	1.92	58	2.17	64	2.16	62	2.26	61	2.42
	40	100	.38	96	.43	88	.62	74	70	1.54	68	1.80	66	2.00	81	1.89	80	2.05	79	2.19
	60	112	.33	109	.38	102	.54	90	86	1.34	84	1.54	82	1.69	98	1.71	96	1.85	94	1.96
2	0	121	.39	116	.45	102	.65	75	68	1.74	62	1.99	58	2.23	54	2.42	51	2.62	48	2.77
	20	129	.35	124	.40	112	.59	87	81	1.57	76	1.80	72	2.00	69	2.20	66	2.36	64	2.54
	30	133	.33	129	.38	117	.56	93	87	1.46	83	1.70	79	1.89	76	2.06	73	2.21	71	2.37
	40	137	.31	133	.36	121	.52	100	94	1.39	90	1.60	86	1.77	83	1.93	81	2.10	79	2.27
	60	145	.27	142	.32	131	.46	112	107	1.21	103	1.38	100	1.54	98	1.71	96	1.85	94	1.96
3	0	152	.33	148	.38	132	.56	101	93	1.59	86	1.84	81	2.08	76	2.27	72	2.46	68	2.62
	20	157	.29	152	.34	139	.51	110	103	1.42	97	1.65	92	1.85	88	2.06	85	2.22	82	2.38
	30	160	.28	155	.32	142	.48	115	108	1.33	103	1.56	98	1.75	94	1.91	91	2.08	88	2.23
	40	163	.26	158	.30	146	.45	121	114	1.26	109	1.47	104	1.64	100	1.79	97	1.95	94	2.08
	60	168	.23	164	.27	153	.40	131	124	1.09	120	1.28	116	1.44	113	1.58	110	1.71	108	1.85

[illegible]

TABLE A-16
FRICTION OF AIR THROUGH VENTO HEATERS

Courtesy of American Radiator Company

Friction Loss—In inches of Water—Due to Air Passing through Vento Stacks
(Measured at 70°)

Regular Section— $4\frac{5}{8}$, 5 and $5\frac{3}{8}$ In. Spacing

Velocity, feet per minute	Spacing of Sec- tions, Inches	1 Stack	2 Stacks	3 Stacks	4 Stacks	5 Stacks	6 Stacks	7 Stacks	8 Stacks
600	$4\frac{5}{8}$	0.022	0.043	0.063	0.084	0.105	0.126	0.147	
	5	.021	.040	.058	.076	.094	.112	.130	0.149
	$5\frac{3}{8}$.019	.034	.049	.064	.079	.094	.109	.124
800	$4\frac{5}{8}$.040	.077	.114	.150	.187	.224	.259	
	5	.037	.070	.103	.135	.167	.200	.232	.265
	$5\frac{3}{8}$.033	.060	.087	.114	.140	.167	.194	.221
1000	$4\frac{5}{8}$.063	.120	.178	.235	.293	.350	.407	
	5	.059	.109	.160	.211	.262	.313	.364	.415
	$5\frac{3}{8}$.052	.094	.136	.178	.220	.262	.304	.346
1200	$4\frac{5}{8}$.090	.172	.255	.337	.420	.502	.584	
	5	.084	.157	.230	.303	.376	.449	.522	.596
	$5\frac{3}{8}$.074	.134	.195	.255	.316	.376	.437	.497
1400	$4\frac{5}{8}$.122	.234	.347	.459	.572	.684	.796	
	5	.115	.214	.314	.414	.513	.612	.712	.813
	$5\frac{3}{8}$.101	.183	.266	.348	.430	.512	.595	.677
1600	$4\frac{5}{8}$.160	.306	.453	.600	.746	.893	1.040	
	5	.150	.280	.410	.540	.670	.800	.930	1.060
	$5\frac{3}{8}$.132	.239	.347	.454	.561	.668	.776	.883

TABLE A-17

SIZES AND DIMENSIONS OF BUFFALO STANDARD FAN SYSTEM HEATER

Length of Section	Section No.	Extreme Height of Heater	Width of Section	Lineal Feet of 1-in. Pipe per Section	Area for Air Passage, Sq Ft	Weight
3' 4 row	1A	3' 4"	8½"	158	5.1	423
	2A	3 10	8½"	178	5.4	454
	3A	4 4	8½"	193	6.1	477
	4A	4 10	8½"	221	6.9	520
	5A	5 4	8½"	249	7.7	564
	6A	5 10	8½"	277	8.5	607
4' 4 row	1B	5' 4"	8½"	320	9.8	739
	2B	5 10	8½"	356	10.8	795
	3B	6 4	8½"	392	11.8	850
	4B	6 10	8½"	428	12.9	906
4' 6" 4 row	1C	5' 10"	8½"	396	12.0	881
	2C	6 4	8½"	436	13 0	943
	3C	6 10	8½"	477	14 0	1006
	4C	7 4	8½"	516	15.0	1066
5' 4 row	1D	6' 4"	8½"	479	14.3	1046
	2D	6 10	8½"	523	15.6	1114
	3D	7 4	8½"	567	16.8	1185
	4D	7 10	8½"	611	17 8	1251
6' 4 row	1E	7' 4"	8½"	670	19 7	1388
	2E	7 10	8½"	722	21.2	1479
	3E	8 4	8½"	774	22.7	1549
	4E	8 10	8½"	826	24.2	1630
7' 2 row	1F	8' 4"	6"	480	27.0	1039
	2F	8 10	6	512	29.0	1089
	3F	9 4	6	544	30.8	1138
	4F	9 10	6	576	32.5	1188

LOSS IN PRESSURE IN BUFFALO HEATERS

Velocity through Clear Area in Feet per Minute	Loss in Ounces per Square Inch per Section of 4 Rows	Loss in Inches of Water per Section of 4 Rows
700	0.027	0.047
800	0.035	0.061
900	0.045	0.078
1000	0.055	0.095
1100	0.067	0.116
1200	0.080	0.138
1300	0.093	0.162
1400	0.104	0.180
1500	0.127	0.220

From Catalogue 197, Buffalo Forge System.

TABLE A-18

DIAMETER OF CIRCULAR PIPES HAVING SAME CARRYING CAPACITY
AS RECTANGULAR DUCTS WITH EQUAL PRESSURE DROPS IN THE
SAME LENGTH

Length	Width												
	3	4	6	9	12	15	18	21	24	27	30	36	48
3	3 3												
4	3 8	4 4											
6	4 6	5 3	6 6										
9	5 5	6 4	8 0	9 9									
12	6 3	7.4	9 2	11 4	13 2								
15	6 9	8 0	10 0	12 6	14.6	16.4							
18	7 4	8 7	11 0	13 6	15.9	16.9	19.7						
21	7 8	9.4	11 6	14 6	17.1	19.4	21 0	23 0					
24	8.3	9 9	12.4	15.5	18.3	20.7	22 7	24 3	26.2				
27	8.6	10.4	13 4	16 5	19 3	21 8	23 8	25 9	27 6	29.6			
30	9 0	10 8	13 8	17.1	20 0	22 9	25 2	27 5	29 0	30 7	32 9		
36	9 7	11 5	14 9	18 6	22 0	24 6	27.4	29 8	31 7	34 1	36 0	39 4	
48	10 9	12 9	16 5	21 2	24 9	28 1	31 8	33 5	36 8	38 8	41 3	45.4	52.6

TABLE A-19

METAL THICKNESS FOR DUCTS

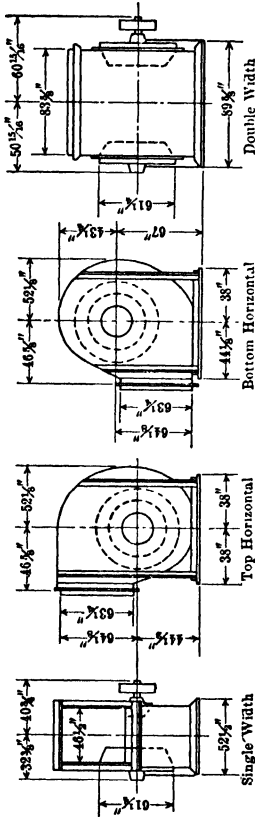
From Fan Engineering, Courtesy Buffalo Forge Co

Gage Number U.S.S.	Actual Thickness, in.	Sizes of Ducts	
		Round Diameter, in.	Rectangular Width, in.
26	$\frac{3}{16}$	6 to 19	4 to 18
24	$\frac{1}{4}$	20 to 29	19 to 30
22	$\frac{1}{32}$	30 to 39	31 to 60
20	$\frac{3}{8}$	40 to 49	61 to 118
18	$\frac{1}{2}$	50	118

TABLE A-20

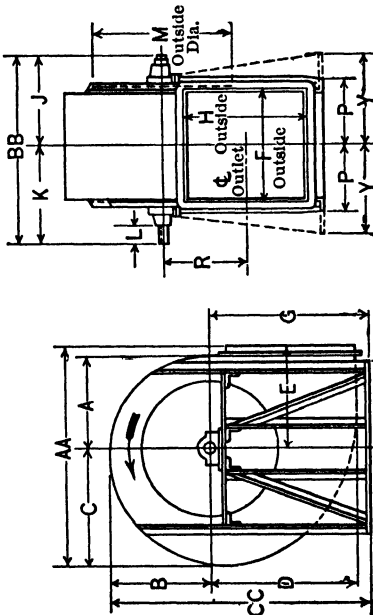
CAPACITIES AND STATIC PRESSURE FOR SIZE 10 BUFFALO LIMIT LOAD CONOIDAL FAN, TYPE CL AT 70°
AND 29.92 IN. BAROMETER. DIMENSIONS OF SAME

Courtesy of Buffalo Forge Company



Outlet Velocity Feet per Minute	Capacity Cubic Feet of Air per Minute	1 1/4 in. S. P.		3/8 in. S. P.		1/2 in. S. P.		5/8 in. S. P.		3/4 in. S. P.		7/8 in. S. P.		1 in. S. P.		1 1/4 in. S. P.		Limit Load	
		Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp	Rpm	Hp
800	16 167	168	0.96	190	1.35	211	1.77	230	2.21	249	2.67	265	3.14	281	3.62	310	4.63	150	0.71
900	18 187	178	1.16	198	1.57	218	2.01	236	2.49	254	2.98	270	3.49	286	4.01	314	5.07	170	1.03
1000	20 208	188	1.39	207	1.82	226	2.29	243	2.80	260	3.32	276	3.86	291	4.42	319	5.58	190	1.44
1100	22 229	199	1.65	217	2.12	234	2.61	251	3.14	267	3.69	282	4.26	297	4.86	324	6.08	210	1.95
1200	24 250	210	1.94	227	2.46	243	2.98	259	3.53	274	4.11	289	4.71	303	5.33	330	6.61	230	2.55
1300	26 271	222	2.29	238	2.83	253	3.39	268	3.97	282	4.57	296	5.19	311	5.85	336	7.19	250	3.28
1400	28 292	234	2.68	249	3.25	264	3.85	278	4.46	291	5.08	305	5.74	319	6.38	343	7.79	270	4.14
1500	30 312	246	3.11	261	3.73	275	4.35	288	4.99	301	5.67	314	6.33	327	7.03	350	8.48	290	5.12
1600	32 333	258	3.61	273	4.25	286	4.85	299	5.60	311	6.29	324	6.99	335	7.70	358	9.20	310	6.12
1700	34 354	270	4.18	285	4.78	297	5.47	309	6.24	322	6.99	334	7.72	345	8.47	367	10.0	330	7.15
1800	36 375	283	4.74	297	5.41	309	6.17	321	6.93	333	7.72	344	8.50	355	9.28	376	10.9	350	8.20
1900	38 396	296	5.41	309	6.17	321	6.93	333	7.72	344	8.50	354	9.23	365	10.1	386	11.8	370	9.00
2000	40 417	309	6.05	321	6.93	333	7.75	344	8.54	355	9.35	365	10.3	376	11.1	396	12.8	390	10.7
2100	42 437	323	6.78	333	7.78	345	8.61	356	9.48	368	10.3	376	11.2	387	12.2	406	13.9	410	14.5
2200	44 458	336	7.54	345	8.61	356	9.48	368	10.5	377	11.4	388	12.3	398	13.2	416	15.1	430	16.7
2300	46 479	350	8.39	359	9.72	369	10.6	380	11.5	389	12.5	398	13.4	409	14.3	427	16.3	450	19.2
2400	48 500	362	9.27	372	10.7	382	11.7	392	12.7	401	13.7	412	14.6	421	15.6	438	17.6	470	21.8
2500	50 521	375	10.2	385	11.8	395	12.9	404	13.8	414	14.9	424	15.9	433	16.9	449	19.0	490	24.8

TABLE A-22
DIMENSIONS OF STURTEVANT SILENT-VANE FANS.
SINGLE WIDTH—SINGLE INLET
Courtesy of B. F. Sturtevant Company



Size Fan	Wheel Diam.	Shaft Ext. Diam.	Keyway			A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	R	Y	AA	BB	CC
			W	D	L																				
57½	15½	1½	¼	¾	3½	9½	10½	13½	16½	10½	11½	18½	14	11½	14½	3½	16½	9½		7½	9		24½	26	29½
60	17	1½	¼	¾	3½	10½	12½	15	18½	11½	12½	20½	15½	13½	15½	4	18½	10½		7½	10		26½	27½	32½
62½	18½	1½	¼	¾	4	11½	13½	16½	20	12½	14½	22½	17½	15½	17½	4½	19½	11½		8½	11		28½	29	35½
65	20½	1½	¼	¾	4½	12½	14½	18	21½	13½	15½	24½	19½	17½	19½	4½	21½	12½		9½	11½		31½	32½	39½
67½	22½	1½	¼	¾	4½	13½	15½	19½	23½	14½	16½	26½	20½	18½	20½	4½	23½	13½		10½	12½		34½	35½	42½
70	24	1½	¼	¾	4½	14½	16½	21½	25½	15½	17½	28½	22½	19½	21½	4½	25½	14½		10½	12½		37½	38½	45½
75	27½	2	¼	¾	5	16½	18½	24½	29½	16½	18½	32½	25½	21½	23½	5	27½	15½		10½	12½		40½	41½	51½
80	31	2½	¼	¾	5½	18½	20½	27½	33½	18½	20½	36½	28½	23½	25½	5½	30½	17½		13½	15½		43½	44½	57½
85	34½	2½	¼	¾	5½	19½	21½	30	36½	20½	22½	39½	30½	25½	27½	5½	32½	19½		13½	15½		46½	47½	61½
90	38	3	¼	¾	6	21½	23½	33½	40½	22½	24½	43½	33½	27½	29½	6	34½	21½		14½	16½		49½	50½	64½
95	41½	3½	¼	¾	6½	22½	24½	35½	43½	23½	25½	46½	34½	28½	30½	6½	36½	22½		14½	16½		52½	53½	68½
100	45	4	¼	¾	7	24½	26½	38½	47½	25½	27½	49½	36½	30½	32½	7	38½	23½		17½	19½		55½	56½	72½
105	49½	4½	¼	¾	7½	26½	28½	41½	51½	27½	29½	53½	38½	32½	34½	7½	40½	24½		17½	19½		58½	59½	76½
115	55½	5	¼	¾	8	28½	30½	44½	55½	29½	31½	57½	40½	34½	36½	8	42½	25½		17½	19½		61½	62½	80½
125	61½	5½	¼	¾	8½	30½	32½	47½	59½	31½	33½	61½	42½	36½	38½	8½	44½	26½		18½	20½		64½	65½	84½
135	68½	6	¼	¾	9	32½	34½	50½	63½	33½	35½	64½	44½	38½	40½	9	46½	27½		18½	20½		67½	68½	88½
145	75½	6½	¼	¾	9½	34½	36½	53½	67½	35½	37½	67½	46½	40½	42½	9½	48½	28½		20½	22½		70½	71½	92½
155	83½	7	¼	¾	10	36½	38½	56½	71½	37½	39½	70½	48½	42½	44½	10	50½	29½		20½	22½		73½	74½	96½
165	90½	7½	¼	¾	10½	38½	40½	59½	75½	39½	41½	73½	50½	44½	46½	10½	52½	30½		22½	24½		76½	77½	100½
175	97½	8	¼	¾	11	40½	42½	62½	80½	41½	43½	76½	52½	46½	48½	11	54½	31½		22½	24½		79½	80½	104½

Dimensions are in inches and approximate. Do not use for construction purposes.

TABLE A-23
SHEET METAL GAGE

Gage No.	12	14	16	18	20	22	24	26	28	30
U. S. S.	0.109	0.078	0.062	0.050	0.038	0.0313	0.0250	0.0188	0.0156	0.0125
B. W. G.	0.109	0.083	0.065	0.049	0.035	0.028	0.022	0.018	0.014	0.012

United States Standard and Birmingham Wire Gage for sheets.

TABLE A-24
CAPACITIES OF SERIES 400 TYPE A NESBITT SYNCRETIZERS
50 PER CENT OUTSIDE AIR

Courtesy of John J. Nesbitt, Inc.

Model and Rating	Outside Temperature	-30°	-20°	-10°	0°	10°	20°	30°	40°	50°	60°	70°
454-A 750 cfm A Anemometer 500 cfm B Standard Air	Final Temp.	454-1	119	121.5	124	126.5	129	131.5	134	136.5	139	141.5
		454-2	98	100.5	103.5	106.5	110	113	116	119	122	125
		454-3	75	78.5	82	85.5	89	93	96	99.5	103.5	107.5
		454-4	64	68	71.5	75.5	79.5	83.5	87	91	95	99
	Total E.D.R.	454-1	224	218	213	207	201	196	190	184	179	173
		454-2	176	171	167	162	159	154	149	145	140	136
		454-3	124	121	118	115	111	108	104	101	99	96
		454-4	99	97	94	92	89	87	84	82	79	77
	Heat for Air to 70° E.D.R.	454-1	113	101	90	79	68	56	45	34	23	12
		454-2										
		454-3										
		454-4										
	Surplus E.D.R.	454-1	111	117	123	128	133	140	145	150	156	161
		454-2	63	70	77	83	91	98	104	111	117	124
		454-3	11	20	28	36	43	52	59	67	76	84
		454-4	-14	-4	4	13	21	31	39	48	56	65
463-A 1000 cfm A Anemometer 750 cfm B Standard Air	Final Temp.	463-1	120	122.5	124.5	127.5	130	132.5	134.5	137.5	139.5	142.5
		463-2	100	103	105.5	109	112	115	118	120.5	124	127
		463-3	75	78.5	82.5	86	89.5	93	96.5	100	104	107.5
		463-4	62.5	66.5	70.5	74.5	78.5	82	86	90	94	98
	Total E.D.R.	463-1	338	329	320	312	304	295	286	278	269	261
		463-2	270	263	256	250	243	236	229	222	216	209
		463-3	186	181	177	172	167	163	157	153	148	143
		463-4	145	141	137	133	130	126	123	119	116	112
	Heat for Air to 70° E.D.R.	463-1	169	152	135	118	101	84	67	50	34	17
		463-2										
		463-3										
		463-4										
	Surplus E.D.R.	463-1	169	177	185	194	203	211	219	228	235	244
		463-2	101	111	121	132	142	152	162	172	181	192
		463-3	17	20	42	64	66	79	90	103	114	126
		463-4	-24	-11	2	15	29	42	56	69	82	95

RECOMMENDED VALVE SIZES FOR SERIES 400 NESBITT SYNCRETIZERS TYPE A FOR
RANGE OF OUTDOOR AIR PERCENTAGES INDICATED

Unit Model	¾" Valve	1" Valve	Unit Model	¾" Valve	1" Valve	1¼" Valve
454-1A	0- 67%	67-100%	472-1A	0- 20%	20-100%
454-2A	0-100%	472-2A	0- 80%	80-100%
454-3A	0-100%	472-3A	0-33 ⅓%	33 ⅓-100%
454-4A	0-100%	472-4A	0-100%
463-1A	0-100%	0-100%	476-1A	0- 10%	10-100%
463-2A	0- 25%	25-100%	476-2A	0- 50%	50-100%
463-3A	0-100%	476-3A	0- 10%	10-100%
463-4A	0-100%	476-4A	0- 67%	67-100%

Steam connections for all feed valves shall be at least one size larger than nominal valve sizes. Nesbitt syncretizers are designed for use on vapor or vacuum systems. Each syncretizer must be provided with a thermostatic return trap located at least 6 in. below the radiator return outlet. Traps should be selected on the basis of their ability to pass the condensate shown in the capacity tables.

TABLE A-25
RATINGS AND DIMENSIONS OF UNIT COOLERS OF
BUFFALO FORGE CO.

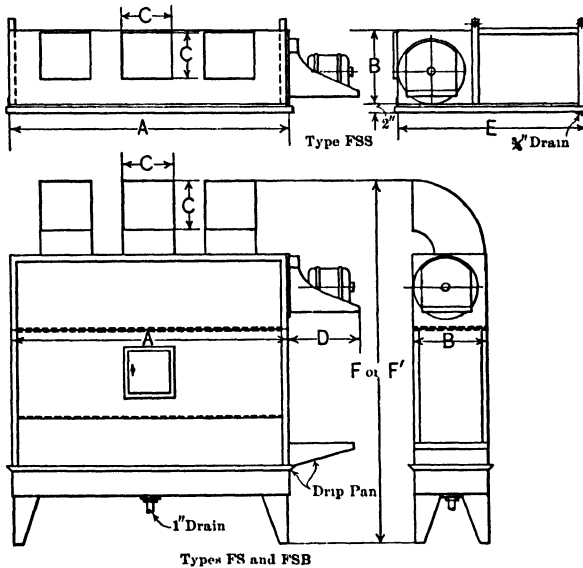
Courtesy of Buffalo Forge Company

RATINGS—FLOOR AND FLAT SUSPENDED UNITS

Size	Rpm	Types FS, FSS, FSB			Types FC and FSC			Motor hp	Water or Brine	
		Cfm	Btu/Hr* per Deg	Temp. Drop per Deg	Cfm	Btu/Hr* per Deg	Temp. per per Deg		Gpm	Pressure Required Lb/Sq In.
254	1150	2850	1900	0.460	2850	2110	0.510	1	35	3
	850	2130	1560	.510	2130	1730	.565	½	35	3
354	1150	4280	2850	.460	4280	3160	.510	1½	40	4
	850	3200	2340	.510	3200	2600	.565	½	40	4
454	1150	5700	3800	.460	5700	4220	.510	2	45	5
	850	4260	3120	.510	4260	3460	.565	¾	45	5
252	1150	3240	1250	.264	3240	1380	.293	1	30	2
	850	2420	1060	.300	2420	1180	.333	½	30	2
352	1150	4860	1860	.264	4860	2070	.293	1½	35	3
	850	3630	1590	.300	3630	1770	.333	½	35	3
452	1150	6480	2480	.264	6480	2760	.293	2	40	4
	850	4840	2130	.300	4840	2360	.333	¾	40	4

* Heat per degree difference between entering air and entering refrigerant (based on 25% latent heat).

DIMENSIONS—FLOOR AND FLAT SUSPENDED UNITS



Size	A	B	C	D	E	F	F'	Size	A	B	C	D	E	F	F'
252	47	18	12 ¾	17	41	90	108	254	47	18	12 ¾	17	50	90	108
352	68	18	12 ¾	17	41	90	108	354	68	18	12 ¾	17	50	90	108
452	86	18	12 ¾	17	41	90	108	454	86	18	12 ¾	17	50	90	108
3152	86	24 ½	16 ¾	24	..	96	114	3154	86	24 ½	16 ¾	24	..	96	114
3552	86	37	20	27	..	112	130	3554	86	37	20	27	..	112	130

F' is for Type "FSB" height may be decreased by omitting legs or by building unit with front discharge

"FS," "FSB" and "FSS" Units have 2-in. extended nipples on 250, 350 and 450 sizes, 2½-in. nipples on 3150 sizes and two 2-in. nipples on 3550 sizes.

TABLE A-26

HEAT TRANSFER—CONTINUOUS TUBE WATER COILS

Courtesy of Aero-fin Corporation

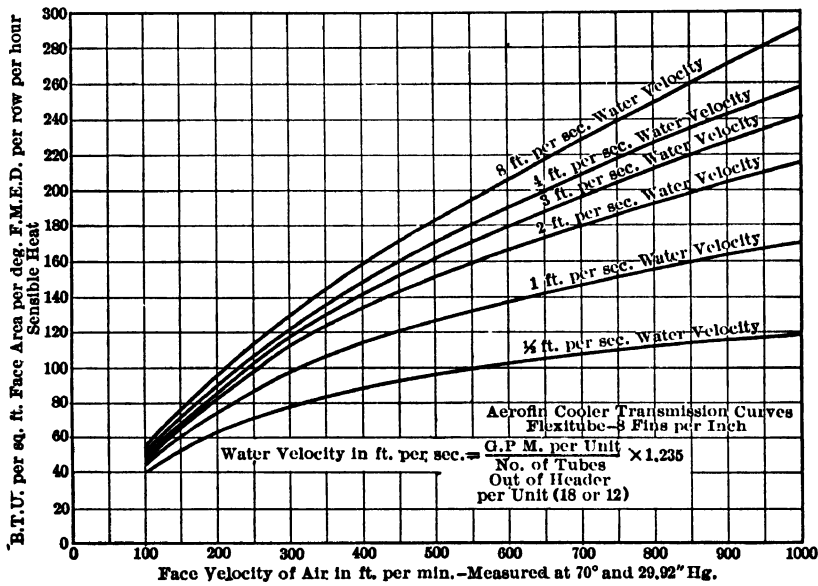


TABLE A-27

DIMENSIONS OF CONTINUOUS TUBE AEROFIN COOLING COILS WITH SENSIBLE HEAT FACTORS. 18 TUBES HIGH—CASING 10 IN. DEEP

Courtesy of Aerofin Corporation

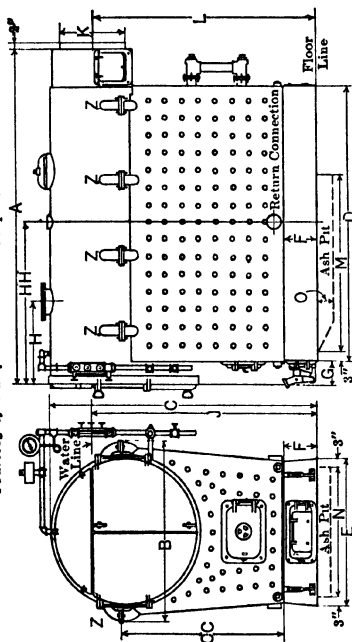
Length of Tubes Feet	Net Face Area	Over-all Dimensions		Length of Tubes Feet	Net Face Area	Over-all Dimensions	
		Across Tubes	Along Tubes			Across Tubes	Along Tubes
2'	4.24	29"	2'-8½"	5'	10.73	20"	5'-8½"
2'-6"	5.33	29"	3'-2½"	5'-6"	11.82	29"	6'-2½"
3'	6.41	29"	3'-8½"	6'	12.9	29"	6'-8½"
3'-6"	7.5	29"	4'-2½"	7'	15.08	29"	7'-8½"
4'	8.58	29"	4'-8½"	8'	17.24	29"	8'-8½"
4'-6"	9.66	29"	5'-2½"	9'	19.41	29"	9'-8½"

FACTORS FOR MULTIPLYING BTU SENSIBLE TRANSFER FROM CHART
WHEN CONDENSING

Percentage Sensible Heat to Total Heat	Ft per Sec Water Velocity					
	0.5	1	2	3	4	8
90	0.940	0.954	0.965	0.970	0.974	0.980
80	.877	.901	.924	.935	.943	.957
70	.806	.842	.876	.893	.907	.928
60	.727	.774	.820	.844	.860	.894
50	.641	.696	.752	.782	.806	.847
40	.543	.605	.670	.706	.735	.787

$$\text{Percentage Sensible Heat} = \frac{(\text{Entering dry bulb} - \text{Leaving dry bulb}) \times 0.241}{(\text{Total heat at entering wet-bulb temperature} - \text{Total heat at leaving wet-bulb temperature})}$$

TABLE A-28
PERFORMANCE AND DIMENSIONS OF PACIFIC BOILERS
Courtesy of Pacific Steel Boiler Corporation



MEASUREMENTS OF PACIFIC LOW WATER LINE BOILERS

Following Measurements also Apply to Smokeless and Oil-fired Types

Catalog Size	Length Over-all	Width Over-all	Height Boiler	Height Firebox Over-all	Length Base	Width Base	Height Base	Shell Overhang	Location Steam Supply	Location Returns	Height Water Line	Diam. Smoke Conn.	Height Center Smoke Conn.	Length for Ash Pit	Width for Ash Pit	Depth for Ash Pit
2281	52 1/4	34	64 1/4	35 1/4	45 1/4	35 1/4	12	3/4	26	Rear	53 1/4	12	51 1/4	30 1/4	29 1/4	
2282	53 1/4	34	64 1/4	35 1/4	51 1/4	35 1/4	12	3/4	26	—	53 1/4	12	51 1/4	45 1/4	29 1/4	
2283	64 1/4	34	64 1/4	35 1/4	61 1/4	30 1/4	12	3/4	32	—	53 1/4	12	51 1/4	51 1/4	29 1/4	
2284	71 1/4	34	64 1/4	35 1/4	67 1/4	30 1/4	12	6	31	—	53 1/4	12	51 1/4	47	24 1/2	
2285	80	39	73 1/4	42	53 1/4	36	12	13 1/2	27	—	59 1/4	14	60	37	30	
2286	86	39	73 1/4	42	58 1/4	36	12	13 1/2	28 1/2	—	59 1/4	14	60	41	30	
2287	92	39	73 1/4	42	64 1/4	36	12	13 1/2	30	—	59 1/4	14	60	45	30	
2288	98	39	73 1/4	42	70 1/4	36	12	13 1/2	31 1/2	—	59 1/4	14	60	45	30	
2289	98	39	73 1/4	42	76 1/4	36	12	13 1/2	33	—	59 1/4	14	60	49	30	
2290	98	39	73 1/4	42	78 1/4	36	12	13 1/2	33 1/2	On Center Line of Water Leg	68 1/4	17	66 1/4	45	34 1/4	
2331	83	44	83	46 1/4	58 1/4	40 1/4	12	15 1/2	31	—	68 1/4	17	66 1/4	49	34 1/4	
2332	95	44	83	46 1/4	70 1/4	40 1/4	12	15 1/2	34	—	68 1/4	17	66 1/4	57	34 1/4	
2333	107	44	83	46 1/4	82 1/4	40 1/4	12	15 1/2	37	—	68 1/4	17	66 1/4	61	34 1/4	
2334	119	44	83	46 1/4	94 1/4	40 1/4	12	15 1/2	40	—	68 1/4	17	66 1/4	61	34 1/4	Not less than 6 in.

STEAM SPECIFICATIONS

Catalog Number	*SHBI Rating	Code Word	Shipping Weight Approx. Lb	Asbestos To Cover Sq Ft	Height Water Line, in.	One Boiler		Two Boiler		Grate Area Sq Ft	Heating Surface Sq Ft	Size Outlet Inches	Size Return Inches
						Diameter Stack, In	Height Stack Feet	Diameter Stack, In	Height Stack Feet				
2261	1360	Fear	2250	54	53 $\frac{3}{4}$	12	45	6.9	97	4	4
2262	1550	Feast	2500	60	53 $\frac{3}{4}$	12	45	8.0	111	4	4
2263	1720	Feat	2650	65	53 $\frac{3}{4}$	12	50	8.0	123	4	4
2264	2100	Fed	3050	76	53 $\frac{3}{4}$	12	55	9.1	150	4	2
2261	2340	Fee	3650	73	59 $\frac{1}{2}$	13	55	9.3	167	5	3
2262	2600	Feel	4100	79	59 $\frac{1}{2}$	13	55	18	60	10.1	186	5	3
2263	2860	Fell	4350	85	59 $\frac{1}{2}$	13	60	18	65	10.9	204	5	3
2264	3110	Felon	4550	91	59 $\frac{1}{2}$	13	60	18	65	10.9	222	5	3
2265	3370	Fen	4800	97	59 $\frac{1}{2}$	13	60	18	65	11.8	241	5	3
2331	3630	Fence	4900	76	68 $\frac{1}{2}$	16	60	21	65	12.5	259	6	3
2332	4350	Fend	5450	89	68 $\frac{1}{2}$	16	65	21	70	13.5	311	6	3
2333	5080	Fern	6100	102	68 $\frac{1}{2}$	16	70	21	75	15.4	363	6	3
2334	5810	Ferro	6700	115	68 $\frac{1}{2}$	16	70	21	75	16.3	415	6	3

* Standard Heating Boiler Institute Code.

48 IN. IDEAL WATER TUBE BOILER—PERFORMANCE DATA—STEAM AND WATER

Number of Boiler	Output in Equivalent Direct Radiation (Sq Ft)													9 625 15 400 2 310 2 460
	Steam (240 Btu)													
	Water (150 Btu)													
	2750 4400	3375 5400	4000 6400	4625 7400	5250 8400	5875 9400	6 500 10 400	7 125 11 400	7 750 12 400	8 375 13 400	9 000 14 400	9 625 15 400	10 250 16 400	
	660	810	960	1110	1260	1410	1 560	1 710	1 860	2 010	2 160	2 310	2 460	
S-4807 W-4807	Time Available Fuel Will Last, Hr *													
Grate Area 21.60 Sq Ft Fuel Capacity 1770 Lb Available Fuel 1328 Lb . . .	17.7	14.2	11.9	10.2	8.9	7.8	7.0	6.3	5.7	5.2	4.7			
	70.5	69.5	68.7	68.0	67.5	66.5	65.5	64.5	63.5	62.5	61.5			
	370	420	470	510	560	610	660	710	750	800	850			
	3.5	4.3	5.2	6.0	6.9	7.9	8.8	9.8	10.8	11.8	13.1			
	0.05	0.08	0.10	0.12	0.16	0.20	0.25	0.28	0.32	0.40	0.44			
	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24			
	70	70	70	70	70	70	70	70	70	70	70			
S-4808 W-4808	Time Available Fuel Will Last, Hr *													
Grate Area 25.20 Sq Ft Fuel Capacity 2065 Lb Available Fuel 1549 Lb	16.8	14.1	12.1	10.6	9.4	8.3	7.5	6.8	6.2	5.7	5.3	4.9		
	70.5	70.0	69.5	69.0	68.5	67.0	66.5	65.5	64.5	63.5	63.0	62.0		
	380	430	460	510	550	590	630	670	710	760	800	840		
	3.7	4.4	5.1	5.8	6.5	7.4	8.2	9.0	9.9	10.8	11.6	12.5		
	0.05	0.08	0.10	0.12	0.13	0.16	0.20	0.25	0.30	0.35	0.40	0.45		
	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24	24 X 24		
	75	75	75	75	75	75	75	75	75	75	75	75		

* Based on Fuel of 12,500 Btu per lb. For economical operation, add piping tax to direct radiation load (equivalent direct radiation).

TABLE A-30
RECOMMENDED MINIMUM CHIMNEY SIZES FOR HEATING BOILERS AND FURNACES*

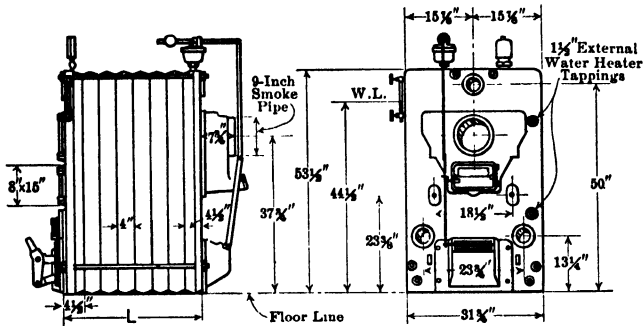
Warm Air Furnace Capacity in Sq In. of Leader Pipe	Steam Boiler Capacity in Sq Ft of Radiation	Hot Water Heater Capacity in Sq Ft of Radiation	Rectangular Flue			Round Flue		Height in Ft Above Grate
			Nominal Dimensions of Fire Clay Lining, in In.	Actual Inside Dimensions of Fire Clay Lining, in In.	Actual Area Sq In.	Inside Diameter of Lining, In.	Actual Area Sq In.	
790	590	973	8½×13	7×11½	81	10	79	35
1000	690	1 140
	900	1 490	13×13	11½×11½	127
	900	1 490	8½×18	6½×16½	110	12	113	40
	1 100	1 820
	1 700	2 800	13×18	11½×16½	183	15	177
	1 940	3 200
	2 130	3 520	18×18	15½×15½	248	45
	2 480	4 090	20×20	17½×17½	298	18	254	50
	3 150	5 200	20	314
	4 300	7 100
	4 600	7 590	20×24	17×21	357	55
	5 000	8 250	24×24	21×21	441	60
	5 570	9 190	24×24 †	576
	5 580	9 200	22	380	65
	5 980	11 500	24	452
	7 270	12 000	24×28 †	672
	8 700	14 400	28×28 †	784
	9 380	15 500
	10 150	16 750	30×30 †	900	27	573
	10 470	17 250	28×32 †	896

* This table is taken from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilating of Buildings (Edition of 1929), by permission.
† Dimensions are for unlined rectangular flues.
This Code and others of the A.S.H.V.E. may be purchased from the Society at its office in New York City for ten cents each.

TABLE A-31
DIMENSION IDEAL BOILERS

Courtesy of American Radiator Company

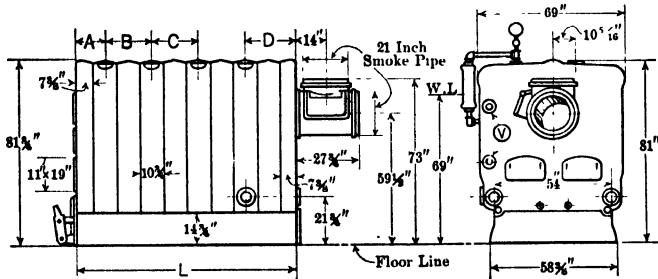
23-IN. IDEAL WATER TUBE BOILER—STEAM OR WATER—ANTHRACITE



Tapped 1 1/2 in. for External Water Heater

Number of Boiler		"L" Length, In.	Outlets Number and Size	Inlets Number and Size	Number of Boiler		"L" Length, In.	Outlets Number and Size	Inlets Number and Size
Steam	Water				Steam	Water			
S-2305	W-2305	21	2-4"	2-4"	S-2308	W-2308	33	2-4"	2-4"
S-2306	W-2306	25	2-4"	2-4"	S-2309	W-2309	37	2-4"	2-4"
S-2307	W-2307	29	2-4"	2-4"	S-2310	W-2310	41	2-4"	2-4"

48-IN. IDEAL WATER TUBE BOILER STEAM OR WATER—ANTHRACITE



Tapped 3 in. on Order for External Water Heater at Points V

Number of Boiler		"L" Length, In.	Steam		Water		Boiler Outlets			
Steam	Water		Outlets Number and Size	Inlets Number and Size	Outlets Number and Size	Inlets Number and Size	A	B	C	D
S-4807	W-4807	68 1/2	3-6"	4-4"	3-6"	4-6"	12 1/2"	21 1/2"	12 1/2"
S-4808	W-4808	79 1/4	3-6"	4-4"	3-6"	4-6"	12 1/2"	32 1/2"	12 1/2"
S-4809	W-4809	90	3-6"	4-4"	3-6"	4-6"	12 1/2"	32 1/2"	12 1/2"
S-4810	W-4810	100 1/2	3-6"	4-4"	3-6"	4-6"	12 1/2"	32 1/2"	12 1/2"
S-4811	W-4811	111 1/2	4-6"	4-4"	4-6"	4-6"	12 1/2"	21 1/2"	21 1/2"	23 1/2"
S-4812	W-4812	122 1/4	4-6"	4-4"	4-6"	4-6"	12 1/2"	21 1/2"	32 1/2"	23 1/2"
S-4813	W-4813	133	4-6"	4-4"	4-6"	4-6"	12 1/2"	32 1/2"	32 1/2"	23 1/2"
S-4814	W-4814	143 1/2	4-6"	4-4"	4-6"	4-6"	12 1/2"	32 1/2"	32 1/2"	23 1/2"

TABLE A-32
DIMENSIONS AND CAPACITIES OF WARM AIR FURNACES
DIMENSIONS AND HEATING CAPACITIES—KELSEY WARM-AIR GENERATORS

Size or Number of Generator	Diameter of Grate and Fire Cylinder	Grate Area, sq in.	Square Feet of Heating Surfaces	Square Feet of Heating Surfaces to Each Square Foot of Grate Surface	Size of Smoke Pipe	Size of Chimney Flue, in.	Inside Diameter of Brick Work if Pit is Used	Size of Cold-air Face in Inches if Air is Taken from Inside	Weight of Generators, "Complete," lb	Number of Average Size Pipes or Rooms	Total Area Heating Pipe Supplied by Each Generator, sq in.
24	24 in.	452	161	51	9 in.	9×12	52 in	24×32	2500	10 to 13	675 to 750
27	27 in.	572½	176	44	9 "	12×12	56 "	30×36	2600	12 to 15	850 to 925
30	30 "	707	211	43	9 "	12×12	60 "	30×48	3124	14 to 19	975 to 1100

DIMENSIONS AND CAPACITIES OF MUELLER FULL FRONT FURNACES

Number of Furnace	Capacity Warm Air Pipes,† sq in.	Capacity Btu Per Hr	Inside Diam Firepot, in	Grate Diam, in	Grate Area, sq in.	Smoke Pipe Diam, in.	Height Floor to Center of Smoke Pipe	Height of Castings Overall, in.	Fire Door Opening, in.	Casing Diam, in.	Approx. Weight, Castings only, lb
F-36	337	45 832	18	15	177	8	41½	48	13¼×10	37	800
F-40	453	61 608	20	18	258	8	41½	50	13¼×10	41	950
F-44	570	71 520	22	21	342	8	45	54	13¼×11	44	1100
F-48 B	673	81 528	24	23	406	9	46	55½	13¼×11	48	1300
F-54 B	755	102 680	27	24	448	9	46	55½	13¼×11	54	1500
*390	975	132 600	30	28	605	10	57½	64½	15½×11¼	58	1850

* Supplied with crank shaker, triangular grates, three-quarter shield, and smooth firepots and feed section only.

† National Warm Air Heating and Air Conditioning Association, certified capacities and measurements.

ANSWERS TO PROBLEMS

CHAPTER II

1. 14,000 cfh; yes (12,000 cfh); 30,000 to 36,000 cfh; 15 cc.
2. D P 56.4° F; R H 44%; abs hum 4.87 gr/cu ft; sp hum 68 gr/lb; T H 29.66 Btu/lb; enthalpy 30.09 Btu/lb; 13.82 cu ft/lb; 0.223 %.
3. Vol 13.33 cu ft/lb; moisture 77.30 gr/lb; enthalpy 26.48 Btu/lb; dry bulb 90° F; wet bulb 70° F; R H 37%; enthalpy 34.19 Btu/lb; vol. 14.12 cu ft/lb; 578 Btu; 7710 Btu.
4. 83.2° F; 74.6° F; 75° F for each.
5. Total 660 Btu/hr person; latent, 330 Btu/hr person; 2210 gr/hr person; 20 gr/lb; 77% R H at 72° F; 53% R H at 84.2° F; yes; 12.2° F; 1470 cfh/person.
6. 7.0° F; more air or cooler air inlet.
7. 952 lb/hr.

CHAPTER III

1. 0.761 Btu/hr; 48° F; 65° F; 0.115° F; 0.031° F; 0.475° F.
2. $U = 0.155$; $U = 0.120$. 3. $U = 0.162$; $k = 0.408$. 4. $U = 0.186$. 5. $U = 0.162$. 6. $U = 0.100$. 7. $U = 0.142$; $U = 0.143$. 8. $U = 0.176$.
9. 20,260 Btu/hr; 14,000 Btu/hr; 5,640 Btu/hr.
10. 2° F; 25° F; 70.3° F; homes, classrooms, offices.
11. 15,800 cfh; yes, 13,650 Mbh.

CHAPTER IV

1. 2-9 sec, 3-tube 26 in. Corto; 2-2 sec, 7 B Peerless wall; 2-11 sec, 20 in. one-tube fantom; 2 No. 518 Arco convector; 4 in. \times 18 in. and 7 in. \times 18 in.; 2-8 sec, 3-tube 26 in. Corto.
2. 40 coils of 8-1½ in. pipes. 13 ft 7½ in. \times 4 ft.
3. 2-7 ft, No. 72 and 2-7 ft, No. 71 flexitube; 6 Vento stacks, 13-60 in. sections 5 in. centers; 7 sections No. 2D Buffalo blast coil units.
4. 9.50 sq ft frontal area.
5. 720 sq ft surfaces; 36 sections, sanitary school radiators.

CHAPTER V

2. Heat demand 13.735 Mbh; Ocp. ht. 1.980 Mbh total, 0.900 Mbh latent; summer load 7.419 Mbh; 0.62 ton of refrigeration; yes, for summer.
3. 66° F; 85.8° F each; 75.9° F; 2730 Mbh; 2960 Mbh; 2619 Mbh; 75,000 cfm.

CHAPTER VI

1. 1 oz in 100 ft; tappings in in.: 1¼, 1¼, 1½, 1¼, 1¼, 1½, 1¼. Rad. branch in in.: 1¼, 1, 1½, 1¼, ¾, 1½, 1¼. Riser in in.: 2½, 2½, 2, 1½, 1½. Riser branch or run-out 2½ in.

2. 0.15 % per 100 ft; tappings in in.: $1\frac{1}{4}$, 1; 1, $\frac{3}{4}$; $1\frac{1}{2}$, $1\frac{1}{4}$; 1, $\frac{3}{4}$; 1, $\frac{3}{4}$; $1\frac{1}{4}$, 1; $1\frac{1}{4}$. Branch in in.: $1\frac{1}{4}$; 1; $1\frac{1}{2}$; $1\frac{1}{4}$; $\frac{3}{4}$; $1\frac{1}{2}$; $1\frac{1}{4}$. B and J branches in in.: $\frac{3}{4}$, $\frac{1}{2}$; $\frac{1}{2}$, $\frac{1}{2}$; 1, $\frac{3}{4}$; $\frac{3}{4}$, $\frac{1}{2}$; 1, $\frac{3}{4}$; $\frac{3}{4}$, $\frac{1}{2}$. St riser in in.: $\frac{3}{4}$, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{1}{2}$, 2. Return riser in in.: $\frac{3}{4}$. Riser runout, 2 in., return runout, $\frac{3}{4}$ in. St riser drip $\frac{3}{4}$ in.

3. St main in in.: 4,4,4,4, 3, $2\frac{1}{2}$; ret. main in in.: $\frac{3}{4}$, $\frac{3}{4}$, 1, 1, $1\frac{1}{4}$, $1\frac{1}{4}$. Drop is given per 100 ft.

4. 335 ft supply; 228 ft return; 0.84 % drop.

5. 2-13 sec 26 in. 3-tube Corto. 6. 6 in. main; 3 in. return. 7. 170 Btu per hr.

8. 18.450 Mbh. 9. 78.5° F.

CHAPTER VII

1. Sq ft surface: 110, 80, 12, 37, 56, 70, 85, 62, 20, 49, 50, 16, total 645. Supply flow main in in.: To A 3, to B 3, to C $2\frac{1}{2}$, to D $2\frac{1}{2}$, to E 2, to J₁ 2, to J₂ 2, to H $1\frac{1}{2}$, to G $1\frac{1}{2}$, to J $1\frac{1}{4}$. Return flow main from A in in.: to B $1\frac{1}{4}$, to C 2, to D 2 to E $2\frac{1}{2}$, to J₁ $2\frac{1}{2}$, to J₂ $2\frac{1}{2}$, to heater 3, from G $1\frac{1}{4}$, from H $1\frac{1}{2}$. 1031 mi.; yes.

2. 7.88 gpm. Supply in order of problem 1: 2, 2, 2, $1\frac{1}{2}$, $1\frac{1}{2}$, $1\frac{1}{4}$, $1\frac{1}{4}$, $\frac{3}{4}$, $\frac{3}{4}$, $\frac{3}{4}$. Return flow as above: 1, 1, $1\frac{1}{4}$, $1\frac{1}{4}$, $1\frac{1}{2}$, 2, 2, $\frac{3}{4}$, $\frac{3}{4}$. 0.00165 hp.

3. $d = 0.508$ in. 4. 8.9 gal; yes. 5. 27 ells.

6. Radiator areas in sq ft on risers: A—59,46; B—62,48; C—48,74; D—51; E—16,26; F—48,32; G—34; H—40,35; I—42,31; J—108,25; flow main 2 in.

7. Use 1 in. pipe, temp. outlet changed from 150° F to 159.0° F.

CHAPTER VIII

1. 5,150,000 cfh; 6750 Mbh, additional; 270% of building loss; 135% of building loss.

2. 0.04 in.; 0.023 oz; 7.4 in. eq. diam.; 136 cfm at 408 fpm.

3. 5.28 in. \times 15.84 in.

4. 1608 fpm; 1.895 lb/sec; 4800 watts; 1.44 in.

5. 567 cfm at 85° F; 551 cfm at 70° F.

6. Cold return air duct 4 in. \times 20 in., hot riser flues 2-3 in. \times 14 in. Surface needed 36 sq ft; use 2 sanitary school pin radiators, 40 sq ft.

CHAPTER IX

1. 42.7° F dry; 20.02 gr/lb. 35.8° F wet.

2. 7 ft, No. 61 flexitube at 1000 fpm.

3. 69° F dry—49° F wet.

4. 2-7 ft, No. 61 flexitube at 563 fpm.

5. 5-6 ft, No. 72 flexitube at 522 fpm. 4-60 in. Vento stacks of 24 sections at 5 in. centers at 1400 fpm. 3 wide and 4 deep, No. 1-D, Buffalo coils at 700 fpm.

6. Mains 47 \times 64, 35 \times 64, 35 \times 47, 30 \times 35; branches 6-12 \times 24, 13 \times 47, 14 \times 35, 14 \times 35; 0.0738 in.; No. 10 Buffalo limit load fan; 251 rpm; 2.85 hp; 108 $\frac{5}{8}$ in. high; 98 $\frac{1}{2}$ in. wide, 73 $\frac{1}{2}$ deep.

7. 158,000 lb/hr at 53° F dry, 46 gr/lb; at least 45,000 lb/hr.

CHAPTER X

1. $2\frac{1}{2}$ ft No. 71 flexitube heater; two No. 472-3A Nesbitt unit heaters could be used as shown by data book.

2. 30,000 cfh fresh air, 167,000 cfh recirculated air saturated and at 63.1° F, $2\frac{1}{2}$ ft Aerofin water cooling coil, 8 tubes deep.

3. Use $2\frac{1}{2}$ ft Aerofin cooling coil, 5 tubes deep.
4. 132 Mbh.
5. 212 tons of refrigeration.

CHAPTER XI

1. Leader diameters in order in in.: 1st floor: 21, $13\frac{1}{2}$, $16\frac{1}{2}$, 21, 19; 2nd floor: $11\frac{1}{2}$, 14, $12\frac{1}{2}$, 14, 14; 3rd floor: 17, $15\frac{1}{2}$, 19, $15\frac{1}{2}$. Using 3.4 lb. coal per sq ft hr 56% eff, 137° F average equiv. reg. temp. Use No. 24 gage or IXXX tin for 1st and 2nd floor, No. 26 or IX tin for 3rd floor. Registers: 1st floor: 20×27 , 15×15 , 16×20 , 20×27 , 20×21 ; 2nd floor: $2-9 \times 9$, $2-10 \times 12$, $2-9 \times 10$, $2-10 \times 12$, $2-10 \times 12$; 3rd floor: $2-12 \times 14$, $2-12 \times 12$, $3-10 \times 14$, $2-12 \times 12$. Stack cannot be placed in 2×4 stud partitions.
2. 38% saving; 63 lb/hr; yes.

CHAPTER XII

1. 4 boilers, 715 boiler hp each; 518 sq ft total; no.
2. 3 No. 30 Kelsey generators, 9 in. smoke pipes. 3 No. 390 Muller furnaces, 10 in. smoke pipes. Code requires a chimney of 240 sq in. 35 ft high.
3. 2 Ideal boilers S-4808; $79\frac{1}{4}$ in. long, 69 in. wide, $81\frac{3}{4}$ in. high; 25.20 sq ft, 21 in. smoke pipe, chimney 8 sq ft total, 75 ft high; same size; 2 Pacific boilers, No. 2293, 86 in. long, 39 in. wide, $72\frac{1}{4}$ in. high; 5 G.E. boilers of larger size.

CHAPTER XIII

1. Diameter, 0.22 in.; 13.21 % absolute pressure; 2. 1270 M lb; 7300 degree days from 65° base.

CHAPTER XIV

1. Diameters in in. given in order: 4, 5, 6, 8, 6, $2\frac{1}{2}$; 3, 4, 5, 6, 5, 2; $2\frac{1}{2}$, 3, 4, 5, 4, $1\frac{1}{2}$. Products: 1030, 1270, 1520, 1960, 1520, 655; 1030, 1330, 1640, 1960, 1640, 700; 1110, 1360, 1750, 2160, 1750, 740; show that there is little difference in heat loss from the various pipes.
2. 34,600 M lb; same; \$3030, \$6060, \$9080, \$12,120, \$12,120, \$9080, \$6060, \$3030, 1.3% for 5 % steam, 37,800 M.
3. 2.00 in. in 100 ft; not over 350 ft; 72 in. radius.

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